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Volume 71 Number 2

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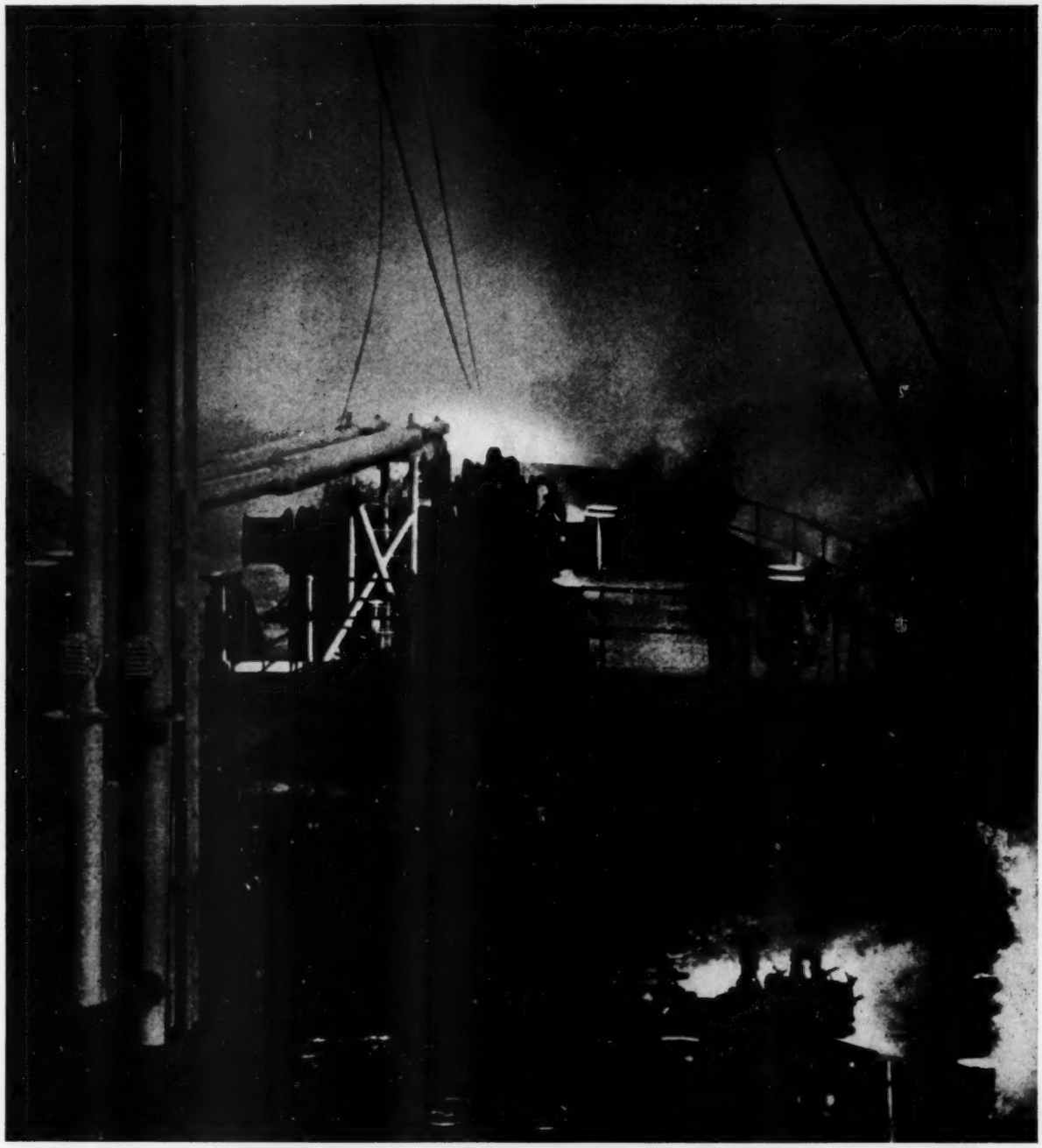

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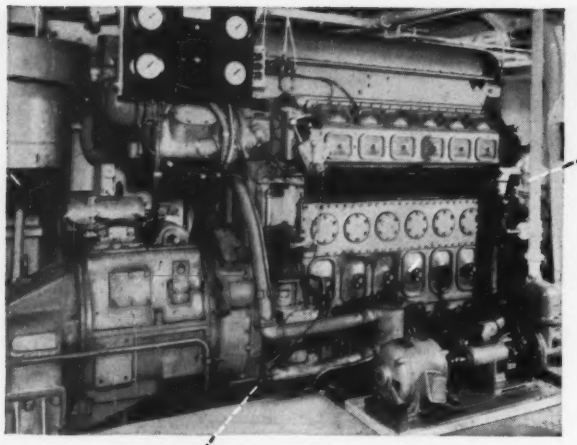

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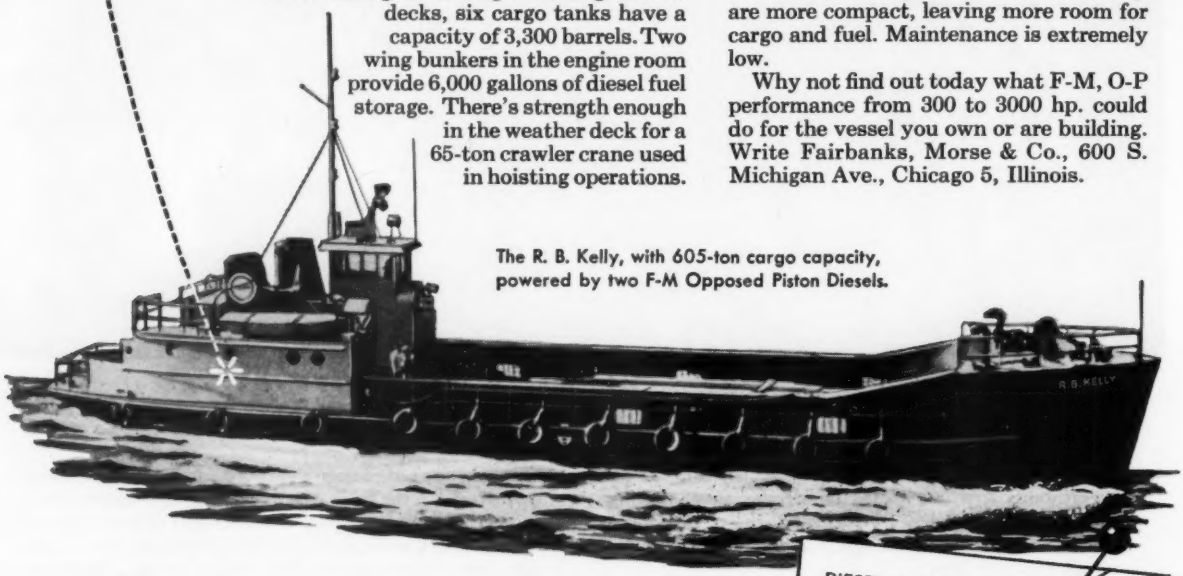

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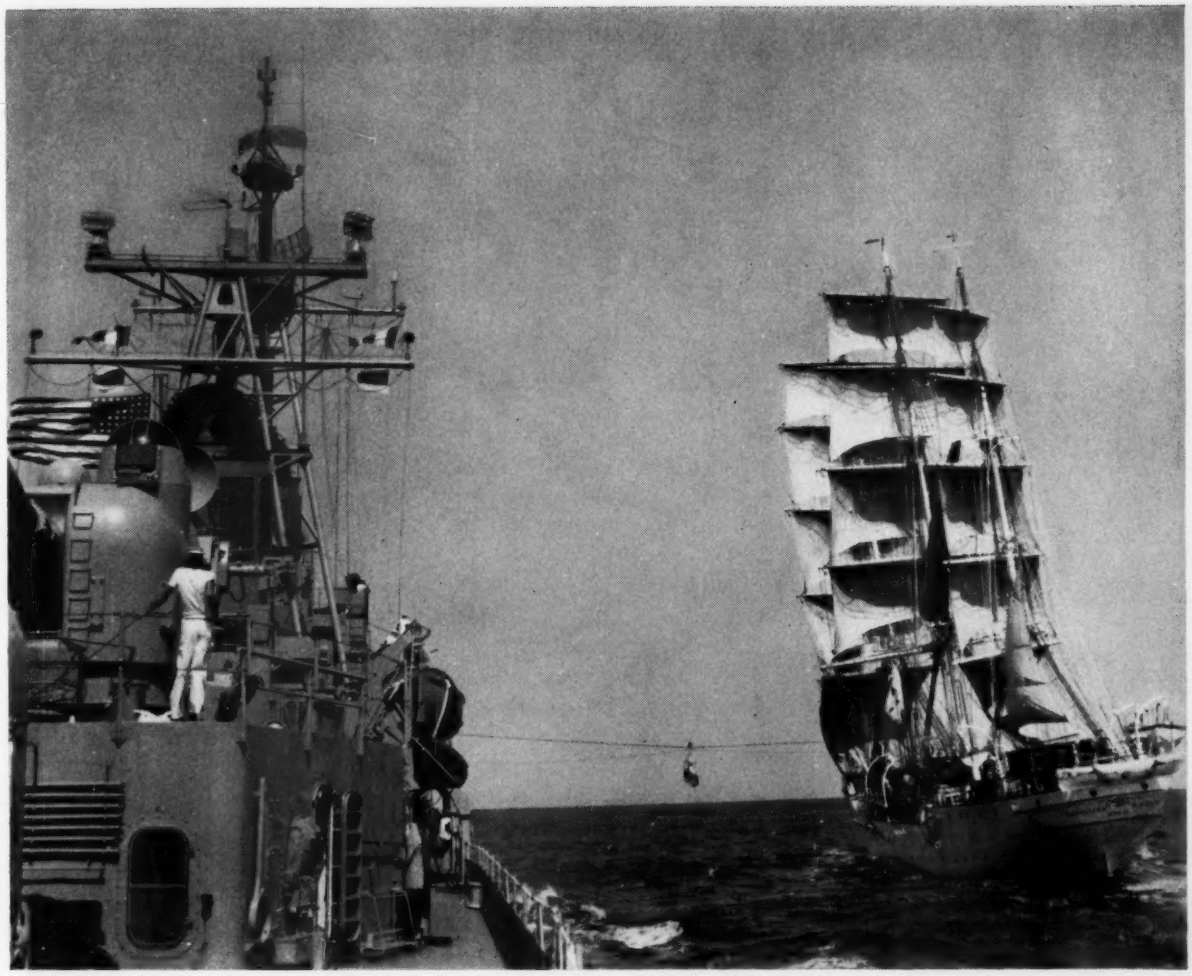

Rendezvous on the high seas — a cadet from the full-rigged ship CHRISTIAN RADICH being transferred to the Bath-built USS FORREST SHERMAN (DD 931) as a fast U.S. Navy task force overtakes the CHRISTIAN RADICH during the filming of Louis de Rochmont's cinemiracle "Windjammer." The CHRISTIAN RADICH hails from Oslo and serves as a training ship for Norway's merchant marine cadets.

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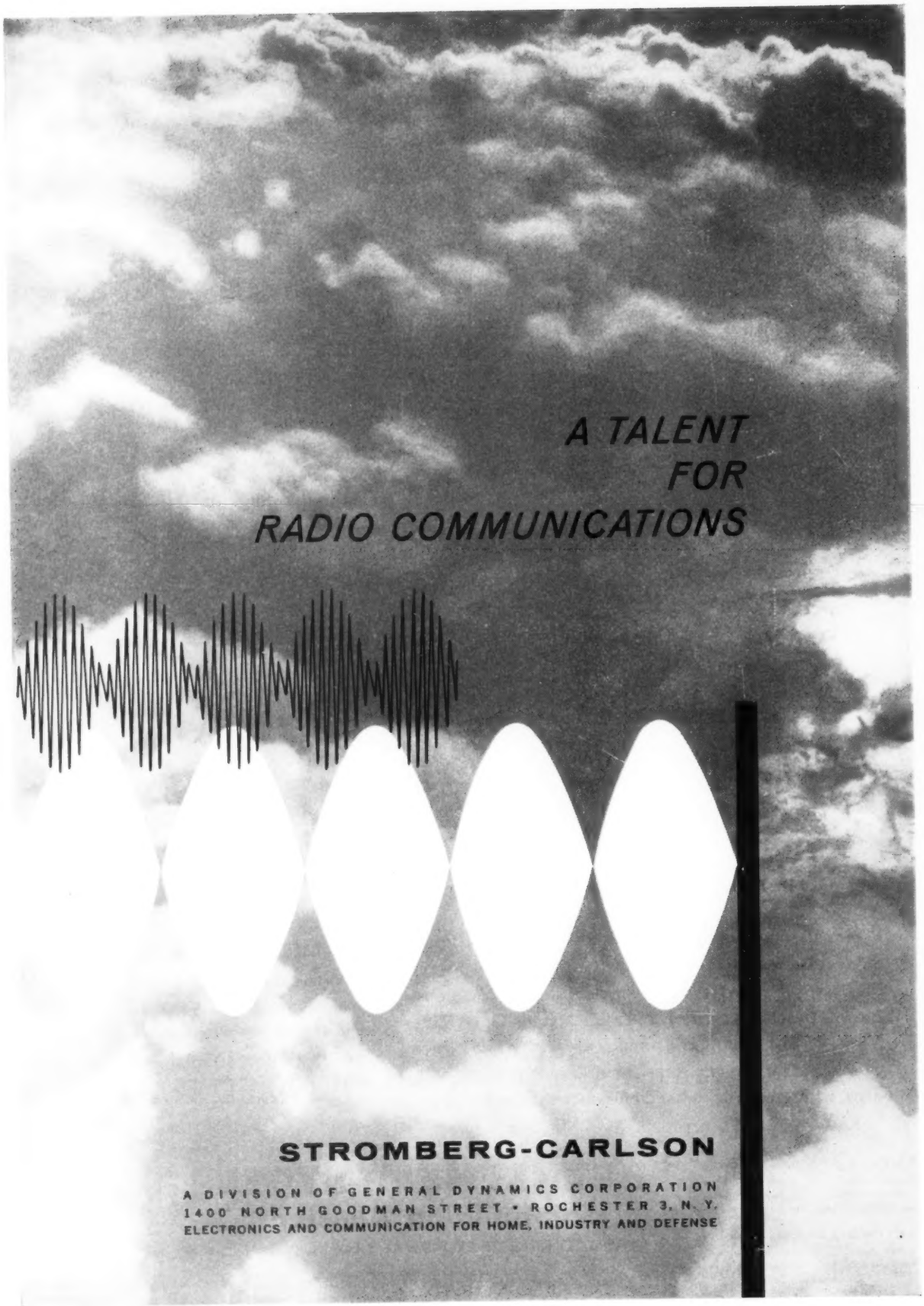
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—from JOURNAL OF THE FRANKLIN INSTITUTE
April, 1959

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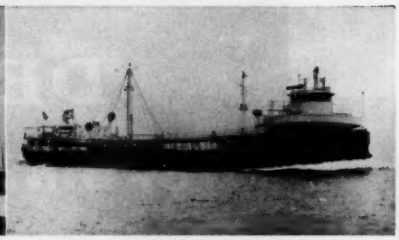
Engines: Napier Deltic
Naval architect: Milne, Gilmore & German
Shipbuilder: Collingwood Shipbuilding Co.

DALZELL TUG NO. D-2, chartered to NH&H RR
Single screw rated 1800 hp at 190 rpm
10' diameter 3-bladed propellers
Engines: Fairbanks-Morse 10 cyl. OP
Naval architect: Thomas D. Bowes
Shipbuilder: RTC Shipbuilding Co., Camden, N. J.



"MARTHA E. ALLEN" *repowered Great Lakes Tanker*
Twin screw, each rated 1600 hp at 225 rpm
9½' diameter 3-bladed propellers
Engines: General Motors
Naval architect: George Drake
Shipbuilder: St. Louis Shipbuilding & Steel Co.

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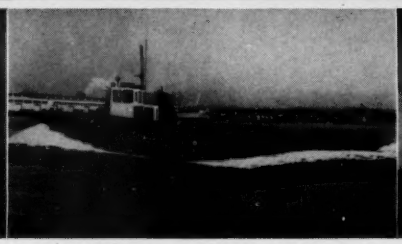
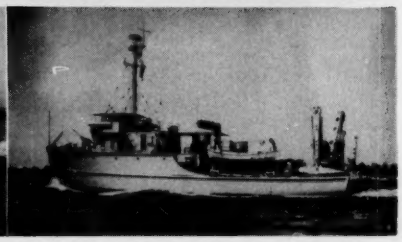
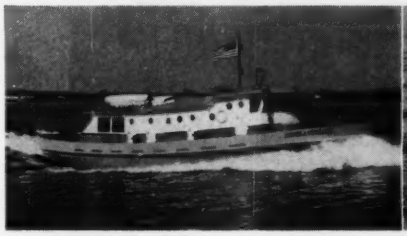
"ALEXANDER T. WOOD" *Ore Carrier*
Single screw, rated 6000 shp at 156 rpm
14' 7" 4-bladed propeller
Engines: Nordberg Mfg. Co.
Naval architect: H. C. Downer Associates
Shipbuilder: Canadian Vickers



M/V TROJAN — CROWLEY LAUNCH & TUGBOAT CO., San Francisco
Single screw, 1000 hp at 425 rpm
76" diameter 3-bladed propeller
Engine: Enterprise
Shipbuilder: Pacific Coast Engineering Co.



U. S. NAVY MINEHUNTER — MHC-43



"JAMES J. VERSLUIS" *ice-breaking tug, City of Chicago*
Single screw, 800 hp at 300 rpm
78" diameter 3-bladed propeller
Engine: Enterprise
Naval architect: A. M. Deering
Shipbuilder: Sturgeon Bay Shipbuilding & Drydock Co.

M/V DELTA CITIES *Twin Screw Tow Boat*
1600 hp each shaft at 240 rpm
9' diameter 3-bladed stainless steel propellers
Engine: Fairbanks-Morse
Shipbuilder: St. Louis Shipbuilding & Steel Co.

RANGER III, *Lake Superior Passenger Vessel*
Operated by Department of Parks
Twin screw each rated 600 hp at 300 rpm
Engine: Kahlenberg
Naval architect: R. A. Stearn
Shipbuilder: Christy Corp.

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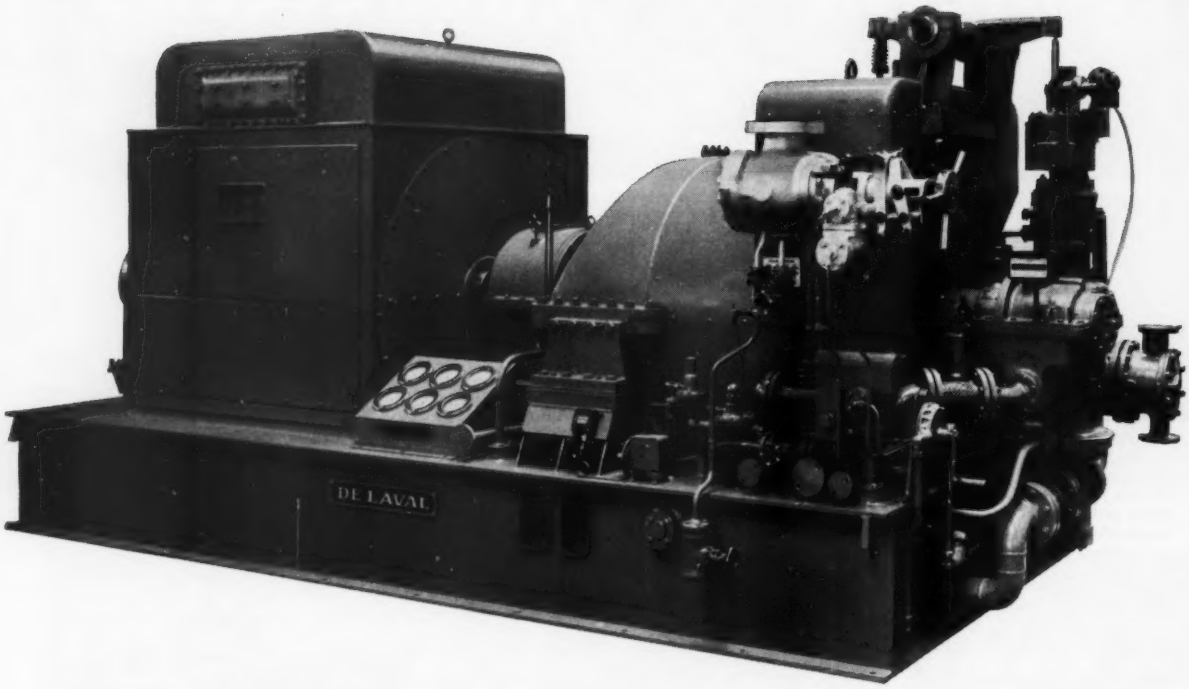
TURBINE GENERATORS

To Supply Electric Power for N/S Savannah



Shown above is one of the two 1500 kw. De Laval condensing turbine generator sets used on the N/S Savannah. The turbines have 8-stages and are of the impulse type.

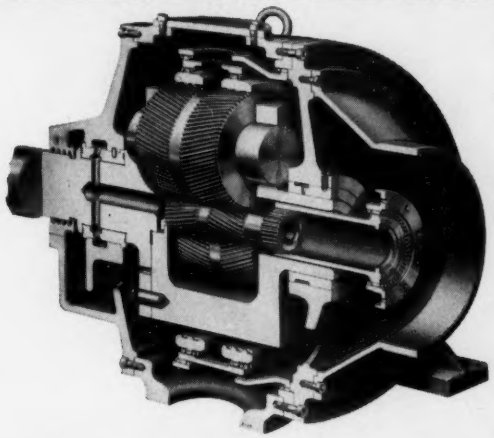
Input speed is 5600 rpm and output speed *via* a size 16 De Laval-Stoeckicht planetary gear is 1200 rpm, at a reduction ratio of 4.67:1.

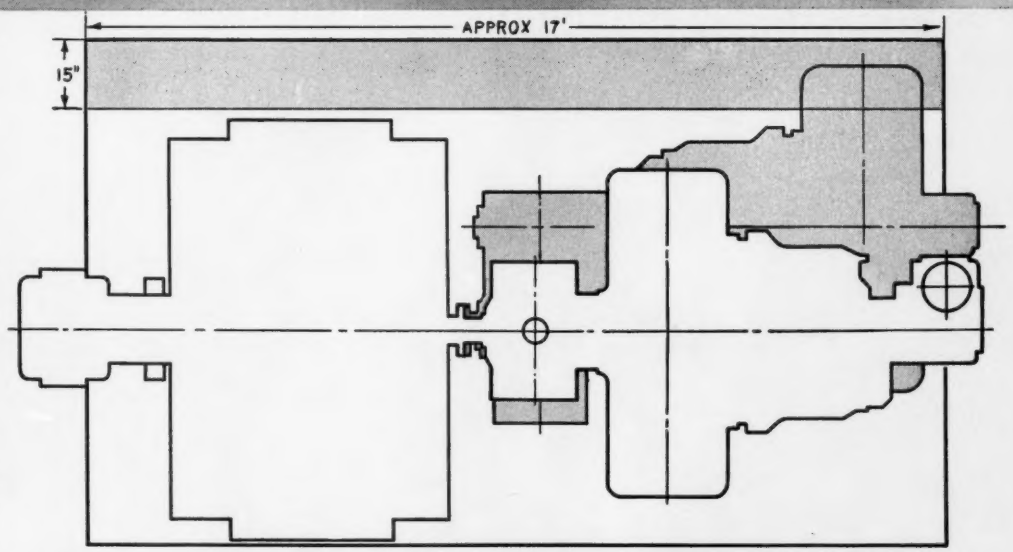
Both these units are designed for continuous rated loads and also for 125% of rated loads for two hours.

• No cast-iron is used. Steel construction provides highest resistance to shock.

DE LAVAL-STOECKICHT
Planetary Gear

This compact planetary gear connecting the steam turbine with the generator weighs only 1750 pounds and is only 22″ in diameter. This is about one-sixth the weight and less than half the size of a conventional double-reduction gear.





Weight of Conventional Gear 6500 lbs. Weight of Stoeckicht Gear 1750 lbs.

Here is a typical plan view of the De Laval turbine generator with the De Laval-Stoeckicht Gear superimposed on a conventional unit. Note how the "in-line" construction of the gear minimizes space requirements. For the N/S Savannah approximately 42 sq. ft. of valuable engine room deck area was saved by the use of the De Laval-Stoeckicht Planetary Gear.

DE LAVAL *Steam Turbine Company*

NOTTINGHAM WAY, TRENTON 2, N.J.

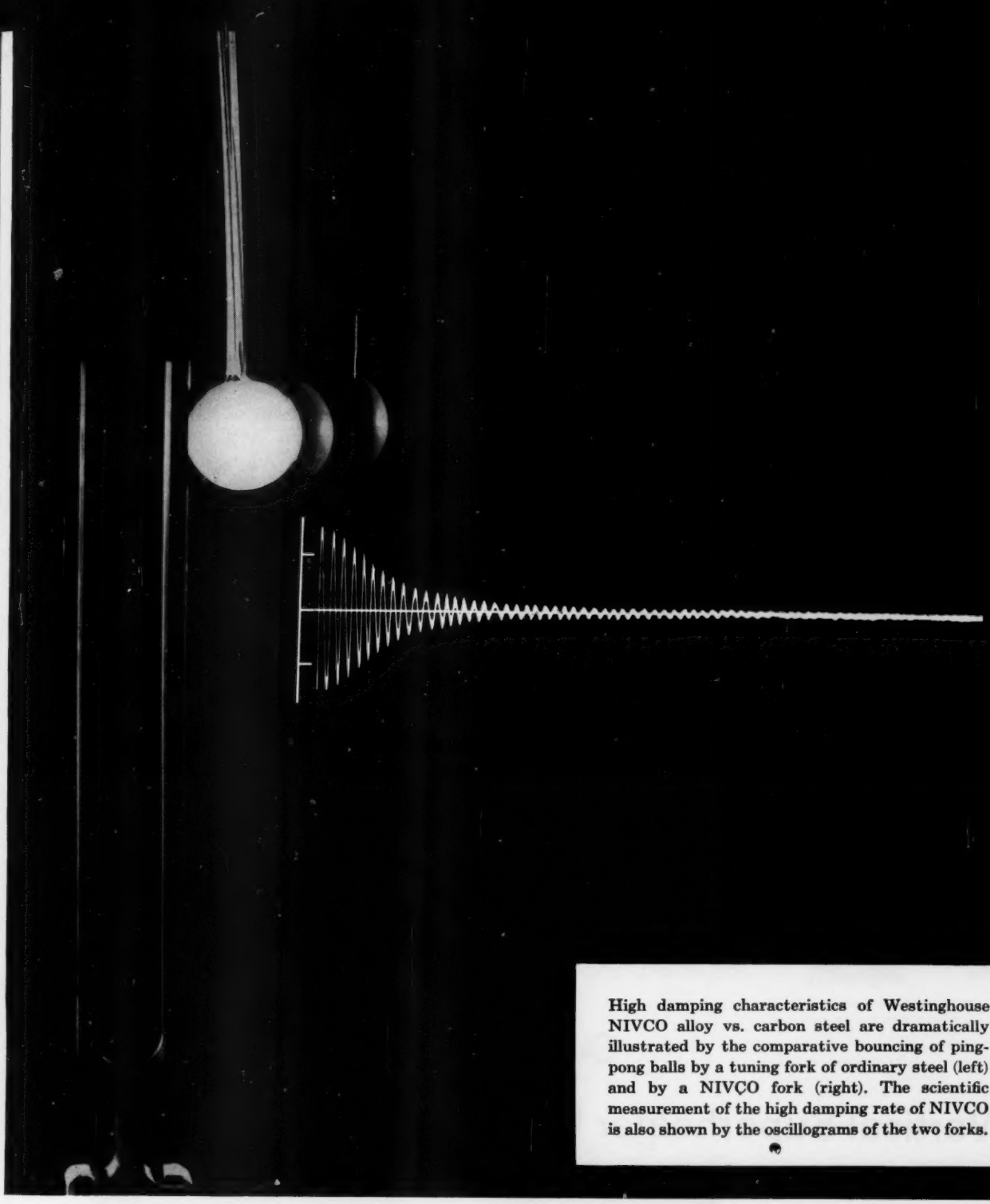
DL-481

WESTINGHOUSE . . . FOREMOST IN MARINE RESEARCH

More Efficient Turbines Soon Possible

Turbines with greatly reduced blade vibration problems may soon be obtainable with a new blading material which is particularly resistant to continuous vibration. This new alloy, "pre-designed" and now being tested by Westinghouse scientists, is one example of how Westinghouse research leads to better, more trouble-free marine propulsion and auxiliary equipment.

Invented specifically for high temperature, high pressure operation, NIVCO, a nickel/cobalt alloy has as much strength at 1200°F as many common alloys have at 900°F. NIVCO's self-damping quality reduces blade vibrations over the wide speed ranges required by marine propulsion. Increased safety factors or more power per unit of machinery is the result.



High damping characteristics of Westinghouse NIVCO alloy vs. carbon steel are dramatically illustrated by the comparative bouncing of ping-pong balls by a tuning fork of ordinary steel (left) and by a NIVCO fork (right). The scientific measurement of the high damping rate of NIVCO is also shown by the oscillograms of the two forks.

Through Applied Metals Research

Other Westinghouse research contributions to the marine industry include:

Thermalastic® insulation for longer-lasting motors and generators;

Hipersil® magnetic alloy for lighter, more efficient motors and transformers.

These and many others make more efficient, low-maintenance marine installations.

Take advantage of these many Westinghouse developments; contact your Westinghouse sales engineer, or write: Westinghouse Electric Corp., 3 Gateway Center, P.O. Box 868, Pittsburgh 30, Pa.

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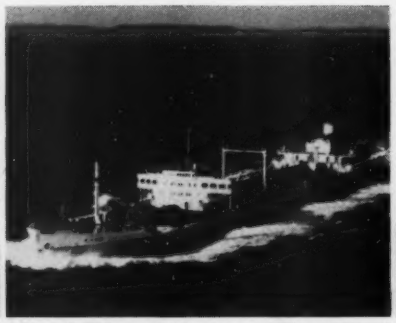
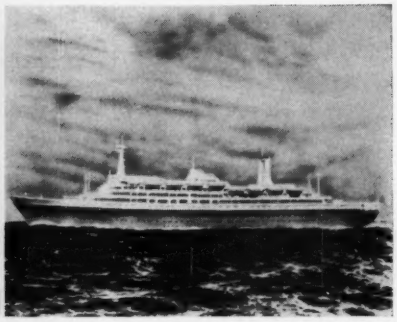
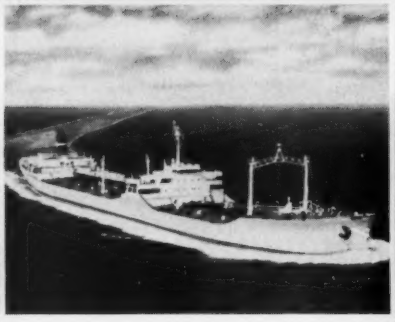
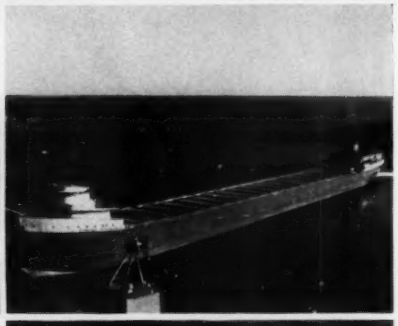
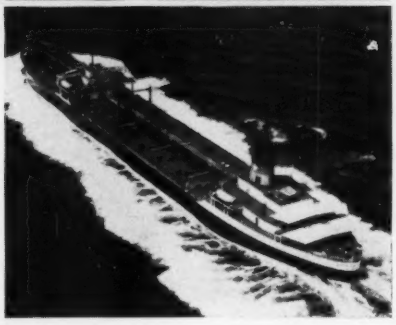



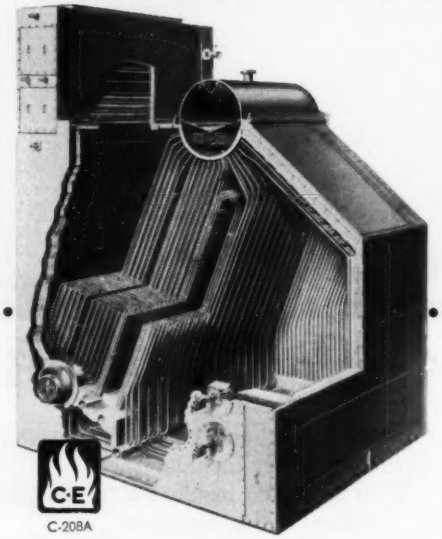





...ships of all types are powered by

C-E V2M

Vertical Superheater Boilers



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CANADA: COMBUSTION ENGINEERING-SUPERHEATER, LTD.

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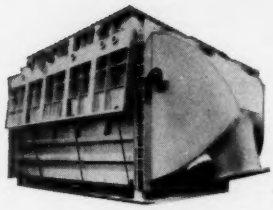
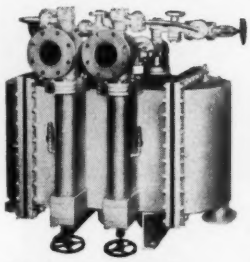
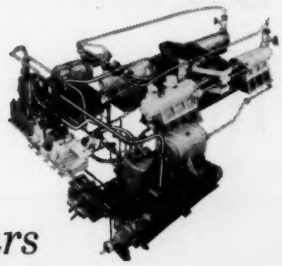
For



Marine Condensers,

C.H. Wheeler Condensers, main and auxiliary, occupy little room, weigh comparatively little, have Center Lane Tube Layout to speed longitudinal steam travel and permit quick penetration of steam to bottom row of tubes.



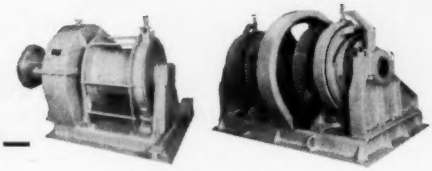


Tubejet® Ejectors, Steering Gears

C.H. Wheeler Tubejet ® Ejectors answer problems raised by today's high boiler pressures and temperatures. They remove large quantities of vapors at high vacuums. Light, compact, dependable, too!

Electro-Hydraulic Steering gears are available in a complete range of sizes for every type vessel, from tug boats to supercarriers. C.H. Wheeler also makes automatic, self-centering, springless telemotors and rotary telemotors.

and Telemotors—



Deck Machinery and

C.H. Wheeler Deck Machinery includes Electric and Hydraulic Winches, Electric Hydraulic Windlasses, Electric Warping Capstans, Boat and Aircraft Cranes, Automatic Tensioning Hoists and Elevators.

High-Capacity Corrosion-Proof Fire Pumps; for

Corrosion-Proof Fire Pumps designed and manufactured by C.H. Wheeler provide maximum gpm at medium and high heads; offer dependable, low-maintenance performance and take little space in the hold.

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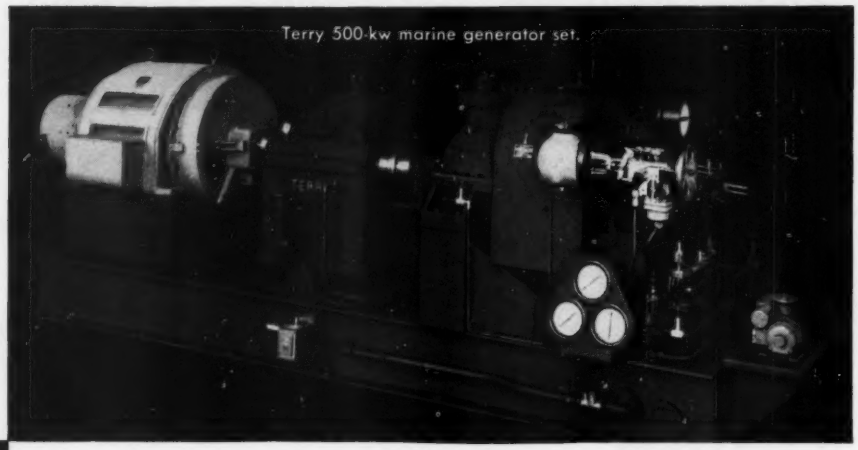
Whenever you see the name C. H. Wheeler on a product, you know it's a quality product

Marine Condensers, Ejectors, Pumps and Auxiliary Machinery • Steam Condensers • Steam Jet Vacuum Equipment • Centrifugal, Axial and Mixed Flow Pumps • Nuclear System Components

Terry steam turbines



Terry 500-kw marine generator set.

Terry solid-wheel turbine with upper half of casing removed.

Dependable, compact, efficient . . . Terry marine turbines are specially designed for driving generators, boiler-feed pumps, fuel-oil pumps, compressors, etc., aboard ship. They are the result of over fifty years of experience in designing equipment for commercial and naval vessels.

Turbines ranging from 5 to 3000 hp are built in the Terry solid-wheel design as well as in axial-flow, single-stage and multi-stage types.

Complete details of turbines for any application will be gladly furnished.

THE TERRY STEAM TURBINE CO.
TERRY SQUARE, HARTFORD 1, CONN.

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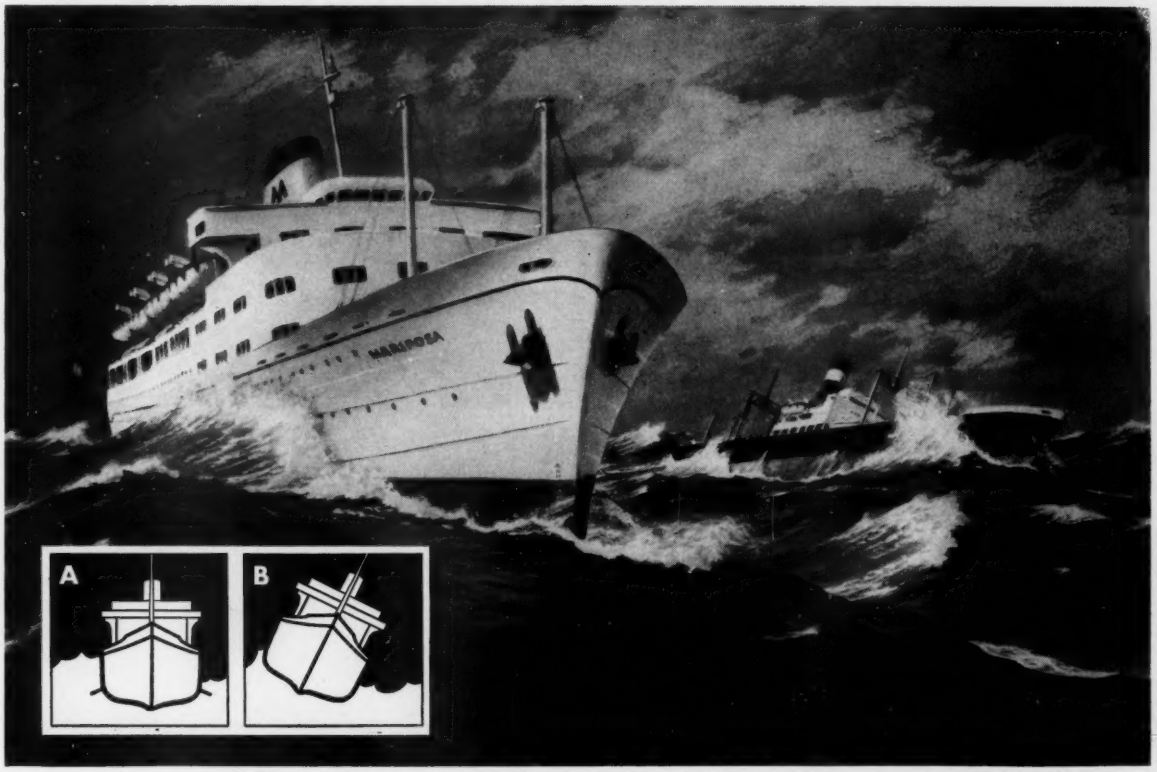
KINGSBURY THRUST and JOURNAL BEARINGS

THE NAVY'S CHOICE FOR 40 YEARS

Kingsbury Machine Works, Inc. KINGSBURY **Frankford, Philadelphia 24, Pa.**



In rough sea, stabilized ship (A) extends fins to halt up to 90% of roll. It holds course without reducing speed. Unstabilized ship (B) rolls and is forced to cut speed. Newport News Shipbuilding and Dry Dock Company, Newport News, Virginia, built the first two commercial Gyrofins which were installed on the Matson liners, *Mariposa* and *Monterey*. Fabrication techniques included extensive use of rugged, corrosion-resisting Monel nickel-copper alloys.

Gyrofin offsets wave action! Monel offsets sea action!

The twin underwater fins of Sperry's new Gyrofin® Ship Stabilizer produce the "lift" which offsets wave action.

But what protects critical parts of the fin assemblies from the corrosive action of sea water? Sperry uses strong, Seagoin'* Monel* nickel-copper alloys.

Take the "hinge pins" for the flap at the trailing edge of each fin. They must take their full share of *3,000 foot-tons* of anti-rolling movement . . . abrasion, too. So they're extra-strong, extra-hard "K" Monel age-hardenable nickel-copper alloy.

Take the bushing where the fin shaft enters the hull. Its spacer rings are Monel nickel-copper alloy. So are the spacer rings at the fin-flap hinge.

Take the rubbing surface of the hydraulic ram that controls the angle of the flap. It must remain true despite abusive metal-to-metal wear . . . despite salt water attack. Welded Monel overlay (deposited on steel with "140" Monel* Electrode) provides the needed wear and corrosion resistance.

Sperry chose Seagoin' Monel alloys for all these Gyrofin parts because of their strength, toughness, hardness and corrosion resistance.

It will *pay you* to choose Monel alloys for marine equipment, too: for propeller and pump shafts . . . for salt water valves and evaporators . . . for metal fittings and fastenings subject to hard use and corrosive attack.

THE INTERNATIONAL NICKEL COMPANY, INC.
67 Wall Street New York 5, N. Y.

MONEL
". . . it's the Seagoin' metal"

INCO TRADE MARK NICKEL ALLOYS

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Dry Dock No. 2

* "We shall build good ships here, at a profit if we can, at a loss if we must—but always good ships."

Collis P. Huntington, founder (1886)

Newly enlarged and modernized for more good* ship repair

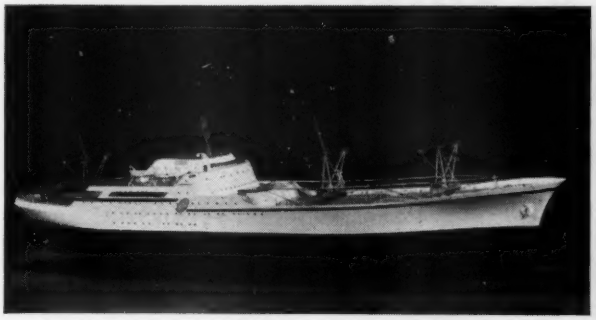
Newport News' Dry Dock 2 was recently enlarged and modernized to provide additional facilities to repair today's larger ships—faster and more economically. This modern graving dock, built for drafts up to 30 feet over its sill, will accommodate ships up to 850 feet in length with a 116-foot beam.

With a total of five graving docks, including two huge docks accommodating ships 960 and 1100 feet long, respectively, Newport News has the facilities to repair ships of any size. As the world's most fully integrated shipyard, it is equipped to do good ship repairs in the shortest possible time.

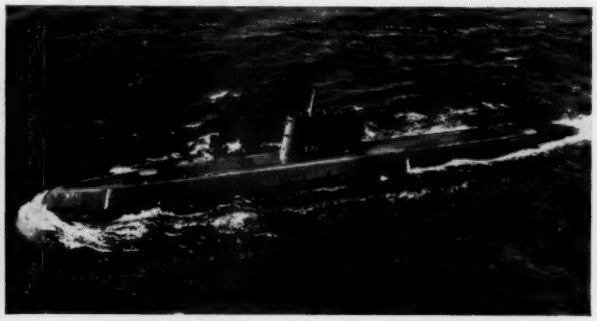
Engineers . . . Desirable positions available at Newport News for Designers and Engineers in many categories. Address inquiries to Employment Manager.

Newport News
Shipbuilding and Dry Dock Company
Newport News, Virginia

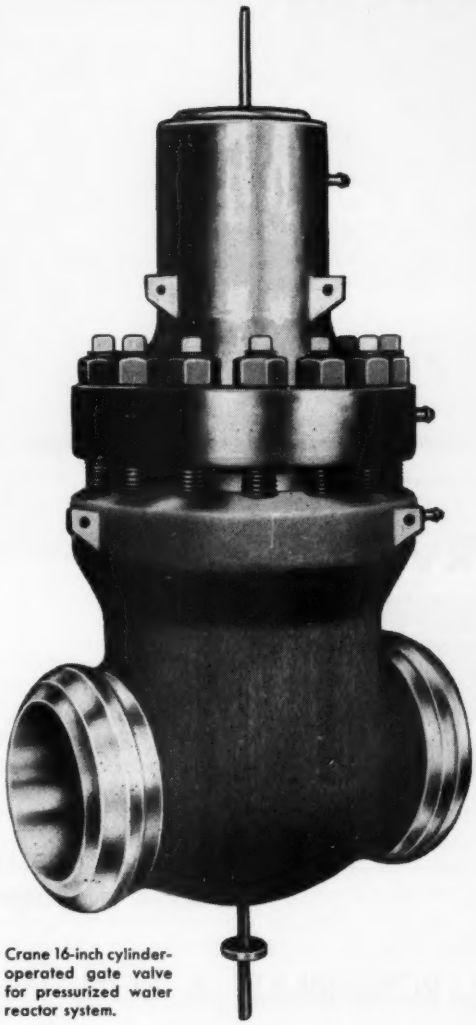
Ever since nuclear power went to sea Crane valves have provided safe controls



N/S Savannah—George G. Sharp, Inc., Designer

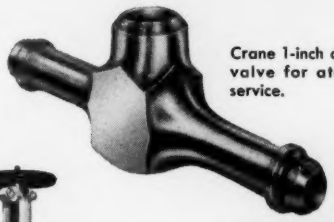


USS Nautilus—Electric Boat Div. of General Dynamics Corp.



Crane 16-inch cylinder-operated gate valve for pressurized water reactor system.

Crane 1-inch capped stem angle valve—3000 psi/500° F.



Crane 1-inch check valve for atomic service.

Crane retractable-expandable, double-disc gate valve with Teflon face. Holds very high vacuum.

Crane globe valve—Monel body, bellows seal for radioactive service.

The daring imagination of naval pioneers who first powered submarines with atomic reactors was matched from the beginning by the resourcefulness, knowledge and skill of Crane engineers and craftsmen.

From their drawing boards, foundries and machines came valves that provided safe flow control of the radioactive materials making nuclear-powered propulsion possible.

Today Crane valves go to sea on reactors and auxiliary service lines on board the *Nautilus* . . . *Seawolf* . . . *Skate* . . . *Sargo* . . . *Skipjack* . . . *Swordfish* . . . *Seadragon* . . . *Triton* . . . and other submarines still being outfitted.

Crane valves will be aboard nuclear-powered surface ships, too: the cruiser *Longbeach* . . . new destroyers and carriers . . . the passenger-cargo ship *N/S Savannah*.

If you are a designer of ships or naval power plants that will use atomic energy as fuel, consult your Crane Representative now for information about valves for nuclear service—or write to address below.

CRANE® **VALVES & FITTINGS**

PIPE • PLUMBING • HEATING • AIR CONDITIONING

Since 1855—Crane Co., General Offices: Chicago 5, Ill. Branches and Wholesalers Serving All Areas

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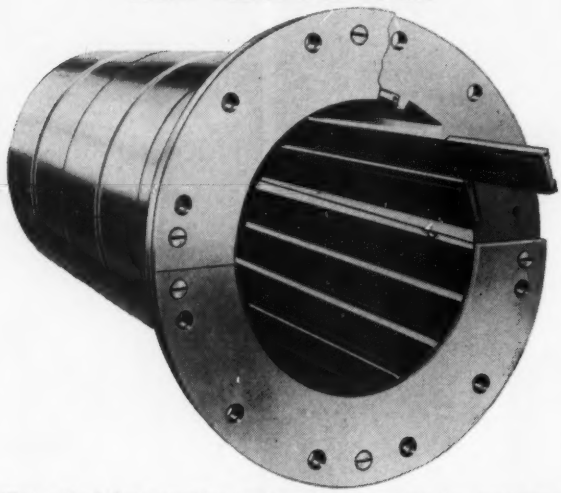
for

Stern Tubes and Struts



Soft rubber bearing surface—efficiently lubricated by water—this bearing far outlasts all hard surface types, protects propeller shafts, reduces vibration. More than pays for itself in extra wear alone. Saves you time, trouble, and upkeep expense.

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Akron, Ohio

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Materials for MARINE SERVICE

Incombustible Joiner Materials • Acoustical Materials

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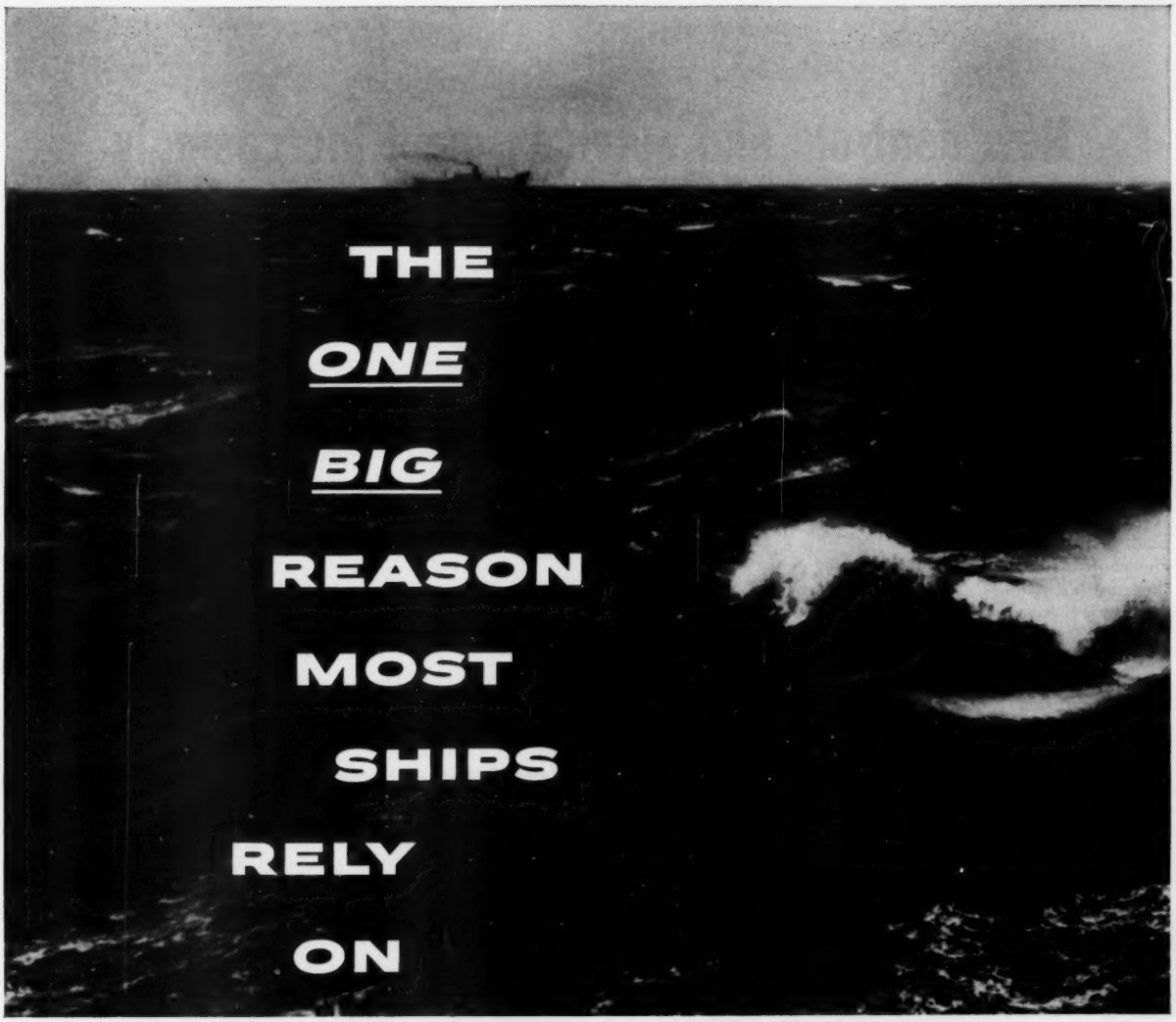

LESLIE FOR CONTROLS

Out here, if a control valve or regulator fails there's no place to go for repair facilities or replacement parts. Delay can mean late arrival, cargo spoilage and loss of profit. Shipowners, operators and builders are aware of this hazard. That's why 95% of America's merchant and naval vessels are Leslie-equipped.

This overwhelming preference for Leslie is based on sound reasons. Leslie reliability means trouble-free service, maximum resistance to corrosion and wear, and lower overall operating cost. Leslie quality is based on nearly 60 years of experience in designing and producing equipment for the marine industry.

Leslie products include *reducing valves* for steam, air and water services; *temperature regulators* for steam and water systems; *pump pressure regulators* for steam driven pumps and turbines; *pilot mechanisms and diaphragm control valves* for liquid level and pressure and temperature control; and Leslie-Tyfon steam and *air whistles*. Write for complete engineering and application data.

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REGULATORS and CONTROLLERS

Leslie Co., 413-A Grant Ave., Lyndhurst, New Jersey

ENGINEERED FOR MARINE SERVICE

Diaphragm control valves

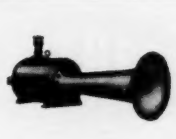
Control pilots for pressure, temperature and level

Pressure reducing valves

Temperature regulators

Pump pressure regulators, Differential pressure regulators

Strainers



Steam and air whistles, automatic whistle controls

New controls and services meet increasingly critical requirements in heat-transfer units

As steam cycles in turbines become more complex—and operating temperatures and pressures rise—the job of the heat-transfer equipment becomes increasingly critical. Nuclear energy plants place tremendous emphasis on continuity of service for even conventional components like condensers. And in many processing plants the cost of shutdowns may range from exorbitant to disastrous.

Assurance that tubes will meet service requirements becomes increasingly important. And The American Brass Company has augmented its normal quality controls with tests and services to help makers of heat-transfer equipment prevent possible trouble before it starts.

Electronic inspection. All tubes are inspected visually one at a time, both internally and externally. However, where greater assurance is required, the tubes—in straight lengths up to 100′—can be inspected electronically by eddy-current equipment.

Hydrostatic testing. As most U-bend tubes are used in applications involving high temperatures and pressures, all of these tubes are tested hydrostatically—after bending—at ASME Code pressures. They can, however, be tested at higher pressures, up to 6000 psi, if the tube size is such that it will withstand the pressure.

Relieving stresses. All U-bend tubes, other than copper, are annealed at the bend area after bending, to eliminate the hazard of stress-corrosion cracking which might occur in service due to stresses that may exist as a result of the bending.

Duplex tubes are widely used in chemical plants and petroleum refineries to meet diverse combinations of corrosive action and/or pressure and temperature. Recently designers have selected them for use in air-removal sections of electric-power-plant condensers where corrosion is very severe.

Broadest service. In helping manufacturers and users of heat-transfer equipment meet day-to-day problems, The American Brass Company has a broad background in the latest developments in heat-transfer equipment's expanding role. And with new mill equipment now in place, it offers the broadest service available in tubes for heat-transfer equipment. For technical assistance in special problems, write: The American Brass Company, Waterbury 20, Conn. In Canada: American Brass Ltd., New Toronto, Ont. 5058B

ANACONDA®

Tubes & Plates for Condensers & Heat Exchangers made by The American Brass Company

---

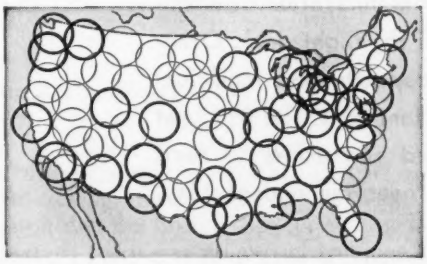
The European shipbuilders, A. B. Gotaverken, have developed a new hatch closure device which has been installed in the Knudsen Line motorship GUDRUN BAKKE, a 10500 DWT cargo ship delivered in December. The hydraulic mechanism which opens and closes split hatch covers 'tween decks is entirely contained in the hatch cover hinge. This compact design allows the hatch covers to be perfectly flush when closed for ease of fork-lift truck operation, and eliminates any encroachment on the cargo space. The mechanism is essentially a screw-jack of simple and rugged construction, which consists of only four working parts for each half cover. A control panel at a convenient location contains the operating valves. The hatches may be completely opened or closed in about 60 seconds, and may be stopped in any position with no special locking arrangement. Extensive shore tests confirmed the reliability and low-maintenace qualities of the design before shipboard application.

—from THE MARINE ENGINEER AND NAVAL ARCHITECT
March, 1959

A Raytheon Flight-Tracker radar mounted on a 75-ft. tower at Indianapolis Municipal Airport keeps an electronic eye on airlanes.

NEW LANDMARKS OF THE JET AGE

Raytheon air traffic radars operating at U. S. airports



To date 26 Raytheon Flight-Tracker radars have been delivered to the C.A.A. for installation at the heavily circled areas. By mid-summer 1959 these long-range units will be installed at a total of 33 airports as part of a nationwide air traffic control network. Light circles indicate future coverage.

Now on the job, the first Raytheon Flight-Tracker radars are already safeguarding commercial and military aircraft. A total of 26 installations will soon be commissioned and linked in a nationwide system of air traffic control.

Flight-Tracker radars work to relieve airlane congestion, reduce stacking, speed schedules. Engineered and produced by Raytheon for the Civil Aeronautics Administration, this new equipment detects and tracks planes—even supersonic jets—day and night, in any weather. It "sees" 4-engine transports up to 200 miles away, at altitudes up to 70,000 feet.

By giving the Jet Age "more sky to fly in", Raytheon radars are contributing to U.S. air transportation progress.

RAYTHEON

Excellence in Electronics

RAYTHEON MANUFACTURING COMPANY, WALTHAM, MASS.

The development of a tiny neutron detector will permit probing all parts of a reactor core to map the neutron flux. This flux distribution, fundamental to proper reactor design, has not been heretofore susceptible to any kind of direct maesurement. Developed by Westinghouse scientists, the detector consists of a tiny solid state diode of silicon or germanium with a thin layer (about a millionth of an ounce) of U-235 on its surface. Encasement of the diode in a spaghetti-sized aluminum tube and wiring it to a suitable amplifier and recorder complete the detector assembly. Fission of an atom of the U-235 by capture of a slow neutron from the reactor fuel results in energetic fission debris which triggers an electric pulse in the diode. Amplified and displayed or recorded, a time history of such events is a measure of the flux at the probe location. The instrument is insensitive to fast neutrons and gamma radiation, a property which enhances its value in providing flux information. Tests have shown that the probe can be used to plot flux variations in dimensional changes as small as one twenty-fifth of an inch.

—from INDUSTRIAL LABORATORIES
March, 1959

Representative of the present program for modernization of World War II type aircraft carriers, USS ORISKANY has recently completed a two year conversion. In order to equip the ship to keep pace with the revolution in carrier aircraft which has taken place since her design, ORISKANY has been equipped with a strengthened flight deck, an angled landing area, and an enclosed "hurricane" bow. Other changes, equally significant, but which alter outward appearances to a lesser degree, are the installation of steam catapults, the arrangement of a functionally designed modular combat information center, and the innovation of aluminum flight deck planking in the touchdown area of the flight deck. Other improvements involved an increase in elevator capacity and modifications to aviation ordnance, fuel and maintenance functions. The 890 foot ship now displaces 44,000 tons when fully loaded and draws 30 feet. Trials and tests are scheduled to be completed near the end of May, when the ship will become a unit of the fleet.

—from BUREAU OF SHIPS JOURNAL
May, 1959

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COUNCIL OF THE SOCIETY

(Under whose supervision this number is published

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Captain J. E. HAMILTON, USN (Ret.), Secretary-Treasurer

NAVAL ENGINEERING

Definition: NAVAL ENGINEERING is the art and science applied in the design, construction, operation and maintenance of naval ships and their installed equipment.

SECRETARY'S NOTES

The Annual Banquet

On Friday, 1 May 1959, the Society held its forty first banquet. These banquets were not initiated until the Society came of age in 1909 and were not held during either of the World Wars. This is the reason that the Society can count 29 years of its life in which there was no *annual* banquet.

We see no purpose in comparing the 1959 banquet with any of those of the past. Attendance was the biggest that we have ever had at the Statler Hilton Hotel. This size is being held down by the requirement, except for very small parties, that there be at least two members for each non-member guest. Even so, the number is too large to be accommodated for dinner in one room in the only hotel in Washington which offers all of the other facilities which are required for our type of party. We offset this as best we can with a closed circuit T.V. which permits everyone to see and hear the program at the Speaker's table if he wants to. To many this is no handicap because the soul of this meeting is not necessarily at the Speaker's table but in the encountering of old friends and making new ones in all of the many spots where these friends gather. There were about seventy subsidiary parties throughout the hotel before and after dinner and these contributed fully as much to our banquet's success as the main event did.

As usual about 40 per cent of the entire membership of the Society were in a position to attend, and since this was well over two-thirds of the attendance, we feel justified in calling the banquet a family gathering.

At other banquets there have been presentations of one kind or another but this year saw the initiation of the ASNE Award to the outstanding naval engineer of the past year. The presentation of the Society's medal (the only one in existence today) and of a Life Membership in the Society was made to the 1958 selectee,

Mr. Valor C. Smith

Greater details of this are in another section of these notes.

The President, Rear Admiral William A. Dolan, Jr., USN, Assistant Chief of the Bureau of Ships, opened the meeting by introducing Captain J. D. Zimmerman, Chaplain Corps., U. S. Navy to say the invocation. After greeting our guests, Admiral Dolan ordered "seats" and the hotel staff took over. It really was a good dinner—as banquets go, superior.

The Secretary-Treasurer again acted as Toastmaster. He introduced those at the Speaker's table. They are shown and named in the accompanying photograph. He also introduced a few Past-Presidents and others who were with other parties on the floor below the Speaker's table.

After the introductions the program consisted of only two events: the ASNE Award presentation and the Principal Speaker's Address.

The award was made jointly by Past-President, Rear Admiral T. C. Lonnquest, who presented the Society's medal and citation, and by President Rear Admiral W. A. Dolan, Jr., who presented the certificate of Life Membership. The citation of award and a biography of Mr. Valor C. Smith appear hereafter.

In selecting a principal speaker the Society ventured further West than at any previous time in our history. As far as can be recalled our western limit heretofore was Detroit. Since the addition of two new states removes California from the border and places it very near to the geographical center of the United States, this move to the West is fully justified even without the basic justification that the Society's conservatism may now safely acknowledge that the Pacific Coast is included in the major part of the Nation's industrial complex which breeds Naval Engineers and practices Naval Engineering.

Our speaker, who heads Western Gear Corporation, can trace his company's origin to the same year which saw the birth of the American Society of Naval Engineers. Both the Society and the Pacific Tool and Gear Company, Western Gears' original name, were founded in 1888. To survive, as each has so successfully, is patent evidence of the soundness of the bases on which they were established.

More about the Speaker, Mr. Thomas J. Bannan, and his deservedly well received address will be found begining on page 201 following.

Society Finances—1958 Statement

As a part of the Association Notes at the back of this JOURNAL is a copy of the 1958 Financial Statement of the Society as audited by a committee and approved by the Council. The Society operations for the year 1958 showed a net gain for the year of about $950.00. However, clearing the books of an accumulation of bad debts (the books are quite current in this regard now) resulted in a paper loss which nets a drop in the Society's net worth of just under $4000.00. It is planned, in the future, to budget such expected losses so that each year's audit will not be based on any questionable account and the net result will not be a loss.

Society Finances—1959 Budget

At a reglar council meeting, the following budget for 1959 was presented by the Secretary-Treasurer and was approved:

INCOME (ESTIMATES)

Dues		
Regular	$32,600.00	
Junior	200.00	
Advertisement (net)	11,500.00	
Sales (net)	300.00	
Subscriptions (net)		
Contract	13,081.82	
Domestic	1,800.00	
Foreign	4,900.00	
Interest on Investment	1,375.00	
*Banquet profit	1,500.00	
Miscellaneous	None	$67,256.82
EXPENDITURES (ESTIMATED-PLANNED)		
The JOURNAL	$36,000.00	
Salaries	16,060.00	
Retirement	3,300.00	
Awards	572.00	
Rent	3,100.08	
Social Security	316.50	
General Expense	5,000.00	
Depreciation	350.00	$64,698.58
PREDICTED GROSS GAIN		$2,558.24

* This item of "profit on the banquet" may be misleading. Normally the banquet account shows that direct costs were less than banquet income which entirely and solely comes from the "plate" cost. This year it is expected that the cash average will be $1500.00. This becomes a contribution to the Society general income as a partial offset of overhead costs. It is estimated that if we distributed all overhead, which we do not, the banquet's share would be between $4000 and $5000.

Naval Technical Corps

Readers are probably familiar with the recent recommendation of the Franke Board, that all categories of Restricted Officers within the Line of the Navy be eliminated. Under this proposal, present Restricted Line Officers would be given the option of qualifying for unlimited line duty or of transferring to a newly established Naval Technical Corps.

In our JOURNAL pages we previously have traced the history of the present group of Line Officers designated for Engineering Duty. The latest of these

THE SPEAKER'S TABLE

Left to Right: Captain Zimmerman, Chaplain; Admiral Furth, Past-President; Admiral Will, Commander MSTS; Admiral Bowen, Past-President; Admiral Honsinger, Past-President; Admiral Stroop, Chief, Bureau of Ordnance; Mr. Haloday, Asst. to the Secretary of Defense for Guided Missiles; Admiral Wilson, Deputy Chief, Naval Operations; Admiral Richardson, Past-Secretary-Treasurer; Admiral Mumma, Past-President; Mr. McNeil, Assistant Secretary of Defense; Mr. Bannan, Principal Speaker; Admiral Dolan, President; Admiral Mills, Past-President (obscured by lectern); Admiral Sylvester, Past-President; Mr. Smith, 1958 ASNE medalist; Admiral Lonnquest, Past-President; Admiral Howard, Past-President; Admiral Cowart, Past-President; Captain Evans, Past-Secretary-Treasurer; Admiral Williams, Past-President; Admiral Peltier, Chief, Bureau of Yards and Docks; Mr. Blunck, General-Manager Statler Hilto nHotel (Absent when picture was taken); Mr. Herr, Honorary Member.

was Captain Madden's very thorough article which appeared in the February 1954 issue of the JOURNAL. At that time the major changes had occurred in 1899, 1916 and 1939 giving the appearance of a cyclic phenomenon occurring about once in each generation. The Franke Board proposal coming in 1959 sustains the cycle if it leads to adoption.

Apparently the proposal must go through four steps before its adoption is accomplished. The Secretary of the Navy has stated that he proposes to act by 1 May 1959. His action may be known to readers before they read this but we may not be able to report this action in this issue. We assume that following the Secretary of the Navy's action, it will be necessary to proceed in the normal fashion through the Secretary of Defense and the President (Bureau of the Budget) to Congressional action leading to enactment of new legislation. How well the upper echelons are predisposed to act we do not know so we attempt no prophesy as to when the recommendation will become dead or a reality.

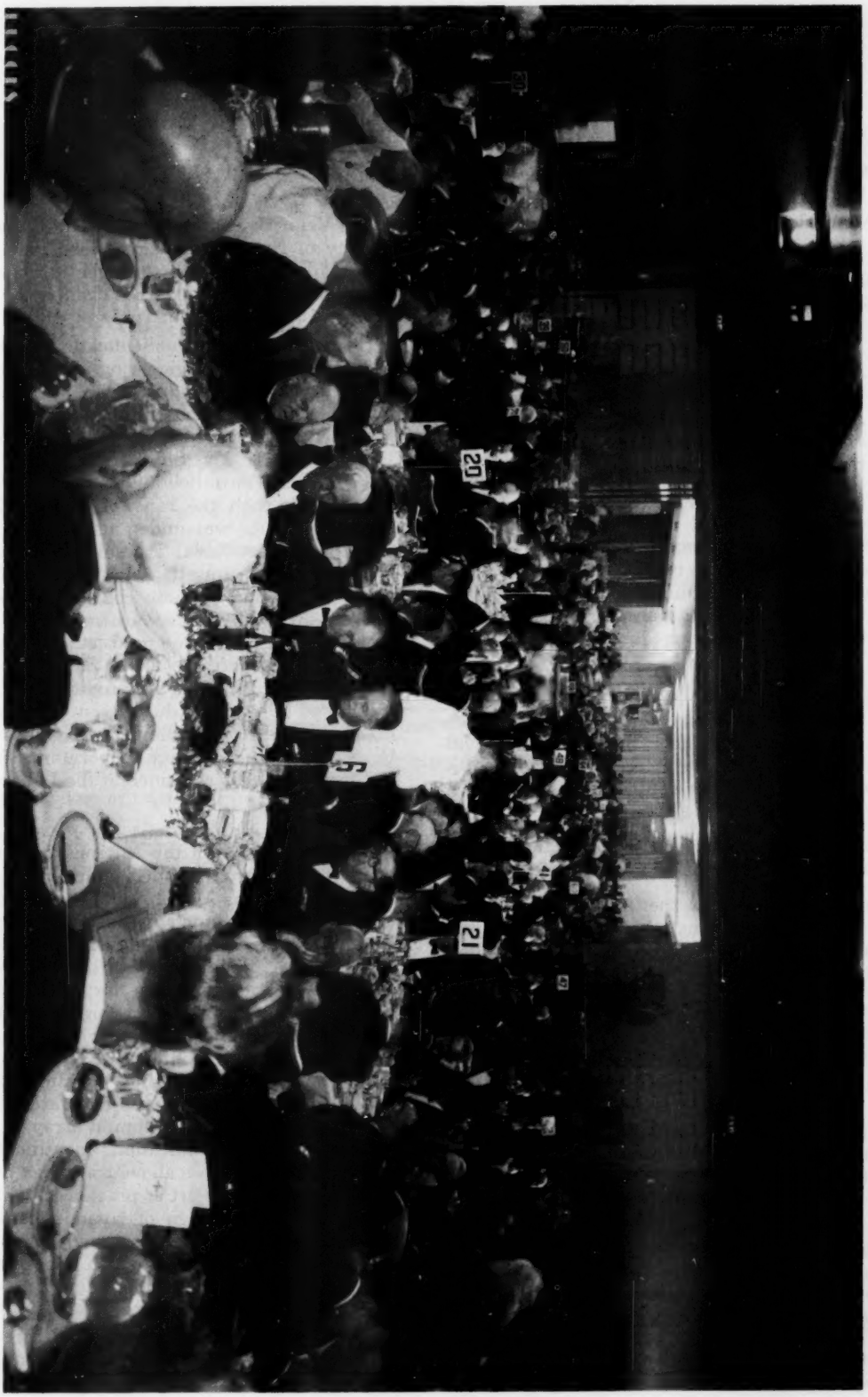
Our sole purpose in writing this is to present facts so that we may keep current in carrying out our self-appointed role as biographer of the Naval Officer Naval Engineer. When the matter is finally disposed of we will have an addendum prepared to bring Captain Madden's paper up to date.

We would like to record a reason for making a change at this time but cannot because we are unaware of any and there has been no authoritative statement which we know of, as to what is wrong to justify the change. We all know the traditional Naval rule "Don't change the set of the sails without a valid reason." If the present "Junior Officer-of-the-Deck" proposes to change the set of the sails he has not yet publicly announced his reason. Presumably he will before the "Captain" (Congress) permits it.

We would also like to report the reasons for doing away with Restricted Line in favor of a new corps. The Board's report includes its reasons for this but since these read much like the reasons for eliminating the Engineer Corps in 1899, for setting up Restricted Line EDOs in 1916, and for eliminating the Construction Corps in 1939 we do not repeat them. There apparently is an opinion which is not affected by reasons and it is the official spread of this opinion which will determine if there is to be a new phase for Commissioned Naval Engineers.

As time goes on this issue will certainly be clarified and we will continue to report the facts as they appear.

FLASH: On 13 May the Secretary of the Navy acted on the Franke Board Report. He has recommended no significant change in the status of line officers designated for engineering duty.



GENERAL VIEW OF THE BANQUET FROM THE SPEAKER'S TABLE

1958 ASNE AWARD

The Committee which was named to consider this matter reported as follows:

April 3, 1959

"Dear Admiral Dolan:

Your letter of 5 February 1959 designated the below named members of the American Society of Naval Engineers as a committee to consider nominations for and to recommend to the Council a nominee to receive the 1958 ASNE medal:

Rear Admiral T. C. Lonnquest, USN (Ret.)
Rear Admiral E. W. Sylvester, USN (Ret.)
Rear Admiral L. V. Honsinger, USN

The criteria for this Award, as understood by your Committee, provide that:

> ". . . annually, if a qualified candidate is found, the Society will award a gold medal and Life Membership in the Society to that naval engineer, military or civilian, who is a citizen of the United States, who in the opinion of the Council has made the most outstanding contribution in the field of Naval Engineering during the preceding year . . ."

and that:

> ". . . the award shall recognize a truly outstanding accomplishment which is in the field of naval engineering and may be a technical writing, invention, design, or other contribution in the field of naval engineering . . ."

Additionally, the Society has adopted the following formal definition for naval engineering:

> "Naval engineering is the art and science applied in the design, construction, operation, and maintenance of naval ships and their installed equipment."

Citations to support the candidacy of several individuals were furnished for the consideration of the Committee by the Secretary of the Society.

Based on the criteria indicated above, and after careful consideration of the accomplishments of each of the candidates as set forth in the supporting citations, the Committee unanimously recommends for the 1958 ASNE Award:

VALOR C. SMITH

A suggested citation, which is appended, summarizes, more specifically, the basis upon which the foregoing nomination is proposed.

Rear Admiral T. C. Lonnquest, USN (Ret.)"

On 8 April, 1959, the Council accepted the above report, concurring in the recommendation and named, as the outstanding Naval Engineer of 1958:

VALOR C. SMITH

Mr. Valor C. Smith, the recipient of the American Society of Naval Engineers award for 1958, which makes him the first person ever to be so honored, was born in San Francisco, California, on 29 December 1919.

He received his higher education from the Polytechnic College of Engineering in Oakland, California from which he graduated in 1941 with a Bachelor of Science in Electrical Engineering degree.

Immediately following the Pearl Harbor attack in December 1941 Mr. Smith enrolled in the Naval Reserve and the following month was ordered to active duty with the rank of Ensign, USNR. His first active duty was under instruction in electronics and radar at Naval Training classes at Harvard and the Massachusetts Institute of Technology. Upon completion of these courses he was retained at MIT for two years as an instructor, being promoted during this time to Lieutenant, USNR. In 1945 he was ordered to sea as electronics officer on the staff of Commander Destroyer Squadron 12. He completed his active duty as a Lieutenant Commander in 1946 and then joined the staff of the U. S. Naval Electronics Laboratory, San Diego, California and has remained at the Laboratory since that time.

At present Mr. Smith is Supervisory Electronic Scientist in charge of the Antenna Systems Section of the Laboratory. Included under his charge is the Ship and Antenna Model Range which has been developed, as an adjunct of the Laboratory, for continuous testing, under simulated conditions, of Navy shipboard antenna systems. In his position and with the model range Mr. Smith has pioneered in a relatively new art, now well recognized and descriptively termed "Electronic Architecture." He has been outstandingly influential in the design of nuclear-powered ships, and this influence can be considered to be general and standard from now on for all ships so that an optimum array of electronics antennas for communication, identification, radar and any other purpose above water can be achieved. The influence of the art as practiced by him has also been felt in the design of communication systems for the Arctic and Antarctic and of ship to shore communications systems which are engineered to meet global action demands.

Mr. Smith resides at 3078 Evergreen Street, San Diego, California with his wife and three children: Pamela Anne, 11; Howard Lee, 8; and Kathleen Anne, 4. California can claim the entire family as natives.

THE AMERICAN SOCIETY OF NAVAL ENGINEERS

takes pleasure in presenting the ASNE Award
for 1958
to
MR. VALOR C. SMITH
Supervising Electronic Scientist
Ship Antenna Systems
U. S. Navy Electronics Laboratory
for contributions as set forth in the following

CITATION:

For an exceptionally outstanding contribution in the field of Naval Engineering in the year of 1958 in connection with radio antenna design. Mr. Smith advanced the state of the art of ship antenna design by his original and unique utilization of ship superstructure as antenna radiating elements in connection with naval radio communications. By this application the number of antennas required per ship has been reduced. Ships of the fleet have been provided with antennas possessing omnidirectional radiation patterns and broad band impedance characteristics which represent outstanding advancements in meeting the rigorous requirements of modern fleet communications. The importance of his work is particularly significant in view of the increasing complexity of shipboard communication installations and the technological advances which are compounding the difficulties of warship design and construction. His superior demonstration of initiative, technical imagination, and engineering knowledge in this development have been an inspiration. By this performance and achievement, Mr. Valor C. Smith has distinguished himself as the outstanding Naval Engineer for 1958.

VALOR C. SMITH

Change In By-Laws

The Council has recently approved a proposed change in the By-Laws which will be submitted to the membership for a vote with the 1960-1 ballot in October of this year.

The intent of the proposed change is to make the position of Secretary-Treasurer Council-appointive instead of Membership-elective.

It seems desirable to state for the record that the proposed change has nothing to do with the present Secretary-Treasurer and his unusually long tenure of office. It is really intended to put the Society on a more businesslike basis at whatever time it becomes necessary to replace the present incumbent. The Council feels that the operation of the business of the Society has become too complicated in 1959 (with no moderation in prospect) to make it a fair proposition for the Society or for the man to have this position filled annually through the competition of an election.

Another possibility would be to amend the By-Laws so that only one candidate would be named for the position. This would, in effect, place the appointment in the hands of a Nominating Committee but would retain the form of an election. Rather than so delegating to a Committee, the Council feels that it is the proper body to select the Secretary-Treasurer and to remove him for cause without artificial actions which would merely complicate.

More formal arguments pro and con will be included with the ballot which reaches you later on this year. This note is included so that the membership will have time to fully consider the change.

THE SPEAKER IN ACTION, Mr. McNeil, Admiral Dolan and Admiral Mills appear.

Admiral Dolen presents the Certificate of Life Membership to Valor C. Smith.

Admiral Lonnquest presents the ASNE Medal to Valor C. Smith.

VALOR C. SMITH

THOMAS J. BANNAN

THE NAVAL ENGINEER— HIS PLACE IN THE POLITICAL ECONOMY

Mr. Thomas J. Bannan

A native Westerner, born in San Francisco and officially a resident of Seattle, Washington, since 1929, Thomas J. Bannan is President and Chairman of the Board of Western Gear Corporation, a firm that specializes in the design and manufacture of all types of machinery both for industrial and government use. The company traces its origin to 1888 and presently has six divisions with Executive Offices in Los Angeles County.

Mr. Bannan began his career with the company as an apprentice machinist and later received his engineering degree from the University of Santa Clara. He also was awarded an honorary LL.D. from Seattle University and is presently Chairman of their Board of Regents.

In addition to being an officer or director of numerous companies and financial institutions, he was National Vice-President of the National Association of Manufacturers and a member of their Board of Directors for eight years. He is past president of the American Gear Manufacturers Association, was one of the organizers and the initial president of the Navy League of the Pacific Northwest. He has served as a member of the Advisory Committee on Naval Affairs for the Thirteenth Naval District, and received commendation from the Navy Department for Western Gear's contribution in designing and producing unique machinery for the country's largest ocean cable laying ship.

His belief that business men have a responsibility to the state and the community has led him to take such positions of leadership in many civic, charitable and business assignments. He was a member of the National Citizens Council for Better Schools and has served in an advisory capacity to both the University of Washington and Washington State College.

What may be of particular interest to the A.S.N.E. members is the fact that his companies have been very active in the marine field and in the design and manufacture of marine machinery extending from propulsion gears to deck machinery.

Western Gear designed what has been spectacularly called the "hardest working gear boxes in the world" which were installed on the winners of the unlimited class hydroplanes competing annually in the Gold Cup Races. It also designed and manufactured the propulsion gearing used on the "M/V United States," the largest and most powerful towboat on the inland waterways; for the Navy the steering engines and control equipment for the aircraft carrier "Kittyhawk," the famed Antarctic Icebreakers, the "U.S.S. Glacier" and "Wind" class. In the maritime field, all of the deck machinery on the new cargo ships and the luxury Moore-MacCormack passenger ships, the "S.S. Brasil" and the "S.S. Argentina" came from Western Gear drawing boards.

Editor's Note: The following is the address which Mr. Bannan delivered on the occasion of the Society's 1959 banquet on 1 May 1959 at the Statler-Hilton Hotel, Washington, D.C.

THE NAVAL ENGINEER— HIS PLACE IN THE POLITICAL ECONOMY

I am happy and indeed grateful for this opportunity to address you. Your Society ranks high with industry, and I feel privileged to have been invited to participate. I take it as a tribute to the West, the rim of Christendom, rather than to me personally, that you have asked me to speak. There was a time when I, a boy trodding the hills of San Francisco, thought you were all dudes and statesmen here in Washington. In fact, I am still impressed. I should say that I am non-plussed that you experts, some of the nation's finest engineering and technical brains, are willing to listen to one as inexpert as I. I submit that perhaps a better title for this discourse would have been:

"A WESTERNER SPEAKS TO THE WISE MEN OF THE EAST"

In preparing this address, I did a deal of reading and a lot of reminiscing. I thought it wise to do some research on the significance of naval engineering, to check the part it has played in human history, in the progress of civilization and in the diffusion of human culture. I was amazed, delighted and frankly surprised at the results. Too frequently other men and other industries got the credit; but behind the men who for three thousand years went down to the sea in ships were the men who engineered the sea power.

This term, sea power, to be used accurately should be taken to include not only the navies of the world, but the total strength and interests on the sea, the merchant shipping, the sea-borne trade —in some cases, even the piracy. When related to this broader and more inclusive concept, the history of naval engineering becomes a great factor in world history, the story of the rise and decline of the great maritime nations. Consequently, the profession holds a key to much of our civilization and culture.

Columbus, for instance, had a good press, but behind him were the engineers who designed the 100 ton, 90 foot *Santa Maria* and her smaller sister ships. Christopher Columbus had a fair education in Latin, geography, astronomy and some drafting, but he would have been lost without his engineers. Long before Columbus—in fact, perhaps as early as 1492 B.C.—the Mediterranean basin was the scene of sea battles, piracy and trade. The remarkable civilization of Crete was based on sea power; economic prosperity as well as national defense depended on shipping. And when Crete fell, about 1400 B.C., it was probably due to the failure of her naval engineers, because history tells us this much: some great naval disaster robbed her of sea control.

And we are not without romance: someone designed Cleopatra's barge, on which she entertained Anthony. And she had armament aboard—even if it was only an asp. Undoubtedly the Queen of Sheba's galley was convoyed by contemporary DEs as she sailed to visit at the court of King Solomon. The Phoenicians must have acknowledged their naval engineers as first class citizens—and we hope paid them well!—the way their ships ranged the Mediterranean with fighting ships ready and available, apparently when needed, to defend the trading posts and preserve the trade monopoly. Because Phoenicia lay at the end of the Asiatic caravan routes, her ships diffused the wealth and culture of the Orient and the Nile Valley throughout the Mediterranean basin. The cargo ships which left the Nile delta distributed the arts and culture of Egypt as well as its wheat. Later on, the richest civilization of the ancient world, that of Greece, rose on the foundation stones brought from Egypt, Assyria and Phoenicia. Behind all of this quickening of civilization were men who understood how to build ships to sail and fight on the Mediterranean.

At peace and at war, engineers are constantly behind the scenes; constantly thinking; constantly planning; constantly building for other men to pilot and ride to glory. Behind the dramatic names like galley, trireme, galleon, junk, Viking-craft, dreadnought, cruiser, carrier, clipper and the new atomic submarines, stand naval engineers. Behind Magellan and Nelson, Perry and John Paul Jones, Sir Francis Drake and Barbarossa stood the men who put the decks under them and without whom, glory would have passed them by.

But on this May day in 1959 there is much more to do than reminisce. History is interesting, but the most interesting and, I might add, challenging fifty years of our civilization may be just ahead—nationally and internationally; and transcending the technical, as apparently in the past, will be the economic and the political.

What do I mean? I mean this:

Communism—Russia—Red China—The Middle East — Berlin — Foreign Competition — Inflation — Government Deficits—Big Labor—the Farm Surplus—and Social Irresponsibility—are all symbols of what I mean. Symbols, yes, but also symptoms of the state of affairs today and of the economic and social health of the years to come. If in problems there are opportunities, and I believe there are, we had better get to work.

I am not one to cry, "All is lost," nor do I think we are lost, but we can lose if we fail to recognize our problem-opportunities and if the very keystone of our strength is dissipated, either knowingly or from want of recognition. Arnold Toynbee has pointed out that 19 of 21 notable civilizations have died from within and not from conquest from without.

We engineers, in the analysis of a problem of our own profession, proceed in a logical manner. We proceed from absolutes, from knowns to unknowns and in an endeavor to resolve unknowns, do not ignore experience. Neither do we ignore the reasons for which the knowns are constant. Why, then, do we lose our way when we venture into political and economic fields?

Have we and other responsible people in this nation lost touch with our traditional values and purposes?

Our founding fathers based their revolutionary action on the rights inherent in man as a creature of God. This concept of man, which was set forth in the Declaration of Independence and upon which is based our Constitution and Bill of Rights, is predicated upon man's personal value and human dignity with its correlatives of equality, of opportunity and liberty, coupled with individual initiative and responsibility. Our founding fathers relied upon and demanded moral integrity as the lifeblood which was to circulate through our body politic and infuse perpetual life into our nation.

If today we think in terms of classes or masses instead of the person; if our course as a nation is determined primarily by the set of the Russian rudder and not by our faith and morals and those of our founding fathers, then indeed we are on perilous seas. We are in danger not only as leaders in the free world but as a nation which some day must balance its books with God and man. If the uncommitted third in Asia and Africa interest us only because we wish to blunt Communist expansion, if public acclaim alone prompts us to support our private schools and charities, and if our interest in political life is self-centered and grasping, I assure you that carriers, submarines and missiles will not save us.

Right here let me make one very important point. It should be crystal clear that nothing in this statement inveighs or argues against military strength. Never before has this nation faced such powerful, aggressive, dedicated and ruthless enemies. Their spokesman has promised to bury us. Military defense will close one avenue of advance, and only a moron would suggest that we confront such arrogance and strength with weakness. But defense is more than the Army, the Navy or the Air Force. You can't annihilate an idea with a weapon system—and Communism is an idea. Weapons are useless without the backing of a strong people and a strong economy.

James Tueslow Adams, in his *Epic of America*, stated that this country faced a new political and social era when it settled the last of its frontiers. Other observers have since insisted, and I agree, that the frontiers of science, of education, of medicine and philosophy present more formidable and rewarding frontiers than the wilderness ever did. But in acknowledging these problem-opportunities and advancing on these new frontiers, is it possible that we are straying from some of the fundamental principles of the frontiersman? He was a tough, self-disciplinarian. He had to be to clear the land and take care of his family. A 35-hour week, featherbedding or a limit on his production could never get the job done. It won't today either, but in some quarters we hear shouting for more and more for less and less, and I say, "Such flabbiness can threaten America."

The challenge to our industry, our way of life, yes, our very jobs, from the industry of Western Europe, Japan, Russia and even of China is too apparent to need discussion. Techniques and methods are learned quickly these days, and we in this country do not have a corner either on intelligence or aptitude. To match $2.50 and $3.00 per hour labor costs, plus fringes and automatic escalation against 80 cent, 60 cent, 20 cent and even 5 cent competition seems futile, particularly where there is recognition and acceptance that it is "production that pays the wages" and where people work "half for the present and half for the future."

Even in Communist Russia job advancement today is based on productivity, and incentives for productivity are recognized with personal rewards.

William McChesney Martin, Jr. stated recently a great concern over the growing distrust in the stability of the American dollar among many informed persons abroad. He stated that several things contributed to that distrust. One was that in a comparatively short time American goods were going to be priced out of the world markets. Another was the current twelve billion dollar national deficit and the implication that the richest nation in the world can't live within its means. Traditionally the American dollar has been a symbol of American strength the world over, strength in a world of nations where uneasiness about shaky currency is chronic and persistent. It is likewise a symbol of strength to our own people and to our internal economy. Yet we are subjecting it to a reckless bombardment of

extravagant, irresponsible and frequently self-seeking theories that can only undermine its stability.

Three indisputable facts are obvious:

1) Adequate defense is essential if we are to survive as a nation—if we are to survive even as persons.

2) Adequate industrial productivity is essential if we are to continue to build that defense and consolidate our social and industrial progress.

3) Dollar stability is essential if we are to realize and appreciate the rewards of defense and productivity.

Can we not then, as engineers, find a formula suitably precise to measure almost every action presently related to our political and social problems? Might we not have a set of standards by which we can evaluate our day-to-day attitude? As a beginning, let us suggest that at least three factors be inserted into our equation when we seek the answer as to whether or not we have been presented or propagandized with a fanciful notion or a practical idea. They are:

1) Is the national defense aided?

2) Is the nation's *real* productivity enhanced?

3) Are we contributing to the stabilization of the dollar?

Think those questions over and the more you do, the more you will agree, I suspect, that the sum total of our national economic needs will be satisfied if all three points are satisfied.

You may say that the formula is too materialistic and moral values have been ignored. Stated bluntly and out of context, it may seem so, but you will recall that earlier we made a plea for a return to the principles of our founding fathers, in essence, a plea that we recognize and accept the Judaeo-Christian fundamentals of morality, self-discipline, self-restraint. The formula proposed rests solidly on that foundation. Therein are contained the absolutes, the knowns, without which the logic of our reasoning and the practicability of our conclusions are open to question.

The cult of self-indulgence wants none of this; there must be an easier way, they say, why must we do it the hard way? In discussing human relationships, in seeking social justice, frequently we hear reference to the rights of man. Man does have God-given rights, though material comfort is not one of them. Today we confuse ease with welfare, and what we forget too frequently is that for every right and privilege which man possesses, there is a corresponding duty and responsibility.

If we wish to maintain our nation and its principles, the goals of justice, mercy, and morality must be integrated with our aims of national and economic security. Today, high on the list of duties not to be ignored, either by individuals or groups, are self-discipline and self-restraint.

Discipline and restraint are needed, but never should we be enticed or bamboozled into accepting the kind imposed in conformity to an inflexible party line by a Politburo or a Chinese Commune. Freely and through our voluntary associations, each one of us needs to pour into our American social order a full measure of service, self-sacrifice and honest purpose.

We need to practice the kind of discipline and restraint imposed upon a free person by himself in the exercise of his moral rights. Then having done so, we will have helped save our nation and, certainly, in the process we will have saved ourselves. And as a further reward, if one is needed, men of future ages to whom we will leave this fine heritage will say, as did Churchill, "THAT WAS OUR FINEST HOUR."

COMMANDER S. R. HELLER, JR., USN and H. NORMAN ABRAMSON

HYDROELASTICITY: A NEW NAVAL SCIENCE

THE AUTHORS

Commander Heller, an Engineering Duty Officer of the United States Navy, received his undergraduate education at the University of Michigan in Naval Architecture and Marine Engineering and in Mathematics. Following typical shipyard duty during World War II he received postgraduate instruction at the Massachusetts Institute of Technology leading to the degrees of Naval Engineer and Doctor of Science in Naval Architecture. Since then he has had design responsibilities in the Bureau of Ships, a maintenance assignment with the fleet, and is now Structural Mechanics Officer at the David Taylor Model Basin.

Dr. Abramson received his bachelor's and master's degrees in Aeronautical Engineering from Stanford University during which time he was engaged in aerodynamic research. He then became Associate Professor of Aeronautical Engineering at the Agricultural and Mechanical College of Texas where his interest turned to structural dynamics. With this diversity of interests he earned the degree of Doctor of Philosophy in Applied Mathematics and Engineering Mechanics at the University of Texas. He is now Manager of Engineering Analysis at Southwest Research Institute where his work in dynamics is continuing.

INTRODUCTION

It has recently become quite fashionable, in discussing Naval problems, to use the term *hydroelasticity*. It means many things to many people, but rarely does it mean anything close to the same thing to members of the same group. This word, which is not to be found in any dictionary, was undoubtedly coined by analogy to *aeroelasticity*. Even the term aeroelasticity is itself of dubious lineage despite the very great attention which has been given in recent years to problems falling within its modern accepted definition.

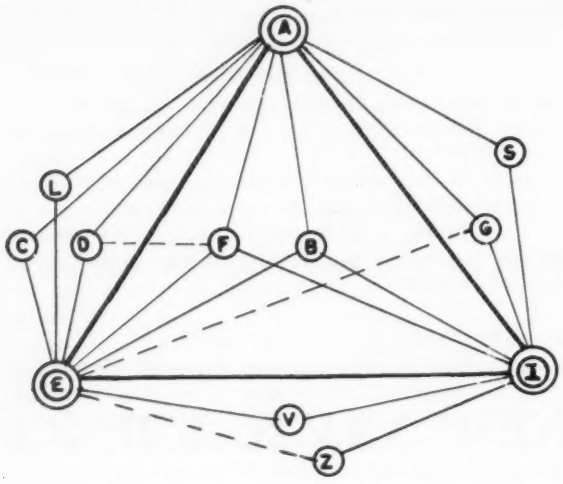
By drawing on aeronautical experience it is proposed to provide the phantasmic concept, *hydroelasticity*, with a skeleton, then to clothe it with substance and body, and finally to circumscribe it with a frame of reference.

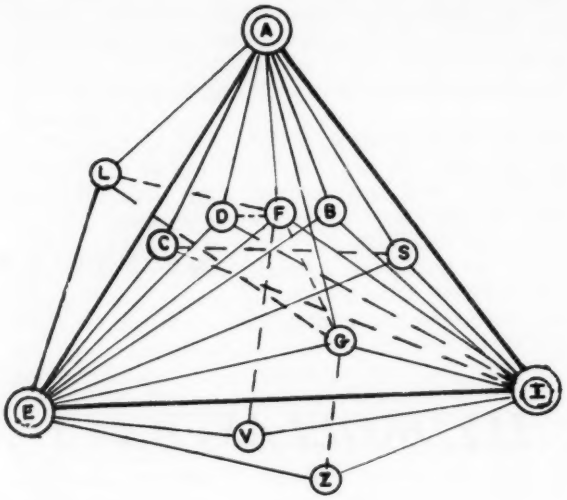
AERONAUTICAL BACKGROUND

Bisplinghoff (1) defines the term *aeroelasticity* as the phenomena involving interactions among inertial, aerodynamic, and elastic forces. He further differentiates between the phenomena involving the three aforementioned types of forces and those involving only the latter two (from which aeroelasticity takes its name). The latter Bisplinghoff calls *static aeroelasticity;* the former *dynamic aeroelasticity*.

Such a definition is entirely compatible with both past and present work in the field and appears to be acceptable to workers therein. As a matter of general information Bisplinghoff's definition seems to be the synthesis of work in the field—the effect not the cause. A definition, however, is not the panacea for our present ills in hydroelasticity. What is needed is limitation of the field.

A step in this direction is the "aeroelastic triangle" developed by Collar (2). Figures 1a and 1b,





Solid Lines: Definite Ties
Dashed Lines: Loose Ties

Figure 1a. Aeroelasticity in 1937.

Figure 1b. Aeroelasticity in 1946.

A: Aerodynamic Forces
E: Elastic Forces
D: Divergence
F: Flutter
I: Inertial Forces
S: Stability

B: Buffeting
C: Control Effectiveness
G: Gusts
L: Load Distribution
V: Mechanical Vibration
Z: Impact

showing the state of aeroelasticity in 1937 and 1946 respectively, have been taken bodily from Reference 2. It may seem strange indeed that, in an attempt to find a limitation of the field, examples of expansion into allied fields are deliberately chosen. From a logical point of view such treatment seems woefully out of line. That this view is superficial can be amply demonstrated by considering that aeroelasticity has been actively pursued as such for some five decades whereas the concept is radically new to the naval architect. Certainly the pride and dignity of the shipbuilding fraternity should not be humbled by drawing on the experience of allied engineering undertakings.

Before Figures 1a and 1b are discarded for purposes of the present discussion, it appears advantageous to describe briefly the various subjects which comprise it. In this summary both Collar (2) and Bisplinghoff (3) have been freely drawn upon for descriptive material.

Buffeting, B. Transient vibrations of components caused by aerodynamic impulses produced by the wake behind wings, nacelles, or other components.

Control effectiveness, C. Influence of elastic deformations of structure on the controllability of an airplane. The complete nullification of the intended effects is called *control reversal.*

Divergence, D. Static instability of a lifting surface, at a critical speed, in which the elasticity of the surface is essential to the instability.

Flutter, F. Dynamic instability of lifting or control surfaces, at a critical speed (or range of speeds), in which the elasticity of the surface(s) is essential to the instability.

Gusts, G. A suddenly applied and removed aerodynamic load.

Load distribution, L. Influence of elastic deformations of a structure on the distribution of areodynamic pressures over it.

Stability, S. The motion of a body in undisturbed flow following a "small" disturbance. *Stability* characterizes the return to the previous position before the disturbance; *neutral stability* characterizes remaining in the disturbed position without any tendency to return to the undisturbed position or to diverge from it; *instability* characterizes the divergence from the disturbed position away from the undisturbed position.

Mechanical vibration, V. A periodic motion of some component.

Impact, Z. A very rapidly applied and removed external load.

NAVAL ARCHITECTURAL COUNTERPARTS

Many of the various subjects which comprise aeroelasticity have their counterparts in the field of naval architecture. In an effort to provide a framework on which to build hydroelasticity, it may be profitable to examine these counterparts briefly.

Although no precise counterpart of buffeting is known to exist in the ship field, a near counterpart, the transient response of structural components produced by pressure fluctuations, turbulence, or wake variations behind appendages or propellers, has long been observed. This will be denoted *dynamic response, DR.*

Up to now ship speeds and control surface dimensions have precluded the instabilities of divergence and flutter. It is likely, however, that as speeds increase and hydrofoil craft obtain acceptance the likelihood of the appearance of flutter and divergence phenomena improves. For example, the sketch in Figure 2 shows that significant changes in configuration are forthcoming which may well lead to operational problems of this nature, not previously encountered. Further, the thin sections required in supercavitating devices have already been observed to exhibit tendencies toward flutter and divergence. To allow for these contingencies the terms *divergence, D,* and *flutter, F,* are retained for the hydroelastic triangle of forces.

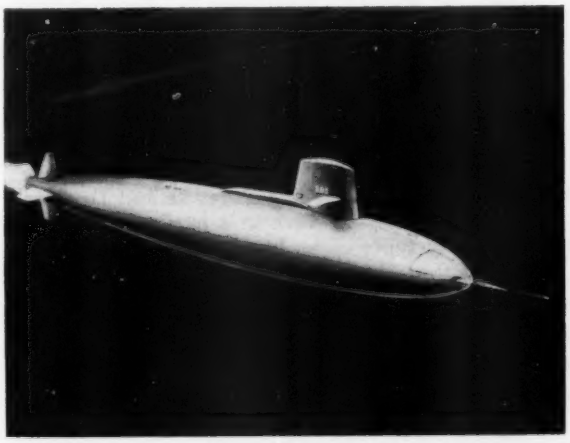


Figure 2. Artist's Conception of USS *Skipjack* (SSN585).

Gusts as such are not experienced by ships, but the steep-fronted shock waves which are characteristic of underwater explosions have similar properties: sudden application and removal by the medium in which immersed. This will be denoted *explosion response, XR.*

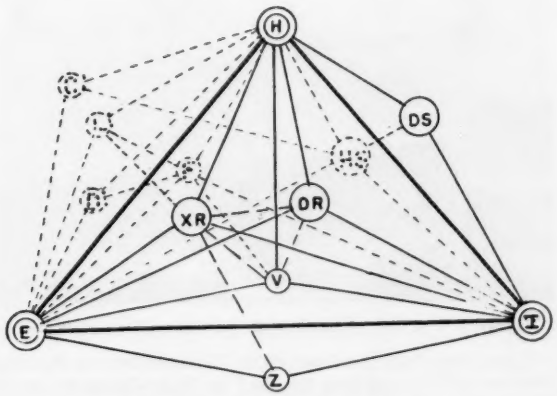
The influence of elastic deformations on load distribution, dynamic stability, and control effectiveness has never been considered by the naval architect heretofore. The latter two will undoubtedly be studied in greater detail as the submerged speed of submarines increases, new configurations are added, and improved facilities such as the Rotating Arm and Maneuvering Basin now under construction at the David Taylor Model Basin become available. These several subjects are called:

Load distribution, L
Dynamic stability, DS
Hydroelastic effects on stability, HS
Control effectiveness, C.

Mechanical vibration, V, has its exact counterpart in ship work. Ship vibration, however, must usually include hydrodynamic forces because the mass of the fluid that is set in motion is not negligible. *Structural-borne noise* is a special manifestation of mechanical vibration. Involved is the transmission of energy from a vibrating source via the structure linking the source to the hull and subsequent radiation by the hull as underwater sound.

Impact, Z, also has its exact counterpart in the ship field resulting from grounding, collision, or enemy attack.

The skeleton of hydroelasticity based on analogy to aeroelasticity is then shown in Figure 3.



Dotted Lines: Probable Areas and Ties
H: Hydrodynamic Forces
HS: Hydroelastic Effects on Stability
DS: Dynamic Stability
DR: Dynamic Response
XR: Explosion Response

Figure 3. Hydroelasticity as derived from Aeroelasticity.

PROBLEMS GENERATED BY FLUID INTERFACE

In many ways the aeronautical engineer has been fortunate since his vehicle is normally operated totally immersed in one fluid medium. He has thus been able to handle as totally separate problems motions and stresses resulting from externally applied loads. In a similar fortunate position is the naval architect concerned only with submarines.

The naval architect who deals with surface ships is not so lucky for his vehicle operates in two fluid media. Consequently, motions, forces, and stresses arising from the action of the interface must be superposed on those already shown in Figure 3.

It is not uncommon for a ship at sea during a storm to be tossed about violently by wave action and even to be pommeled by a wave breaking over its deck into some topside protuberance. Such action might be included under *Impact, Z*, but, since it seems advisable to limit impact to an applied load external to the basic medium, *wave impact* will be dealt with separately.

Similarly, all have seen pictures of small hydroplanes actually leaping out of water only to return with a resounding smack. Not so well-known is the similar action of the long slender naval combatant ship in moderate to heavy seas (see Figure 4). When the bow reenters the water, a somewhat similar impact action results. *Water entry* will also be dealt with separately.

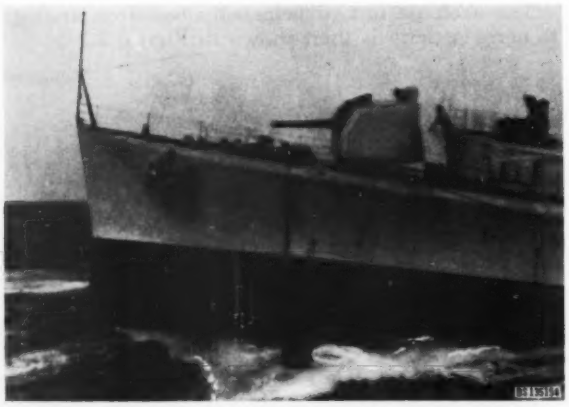


Figure 4. USCGC *Unimak* with Forefoot Clear of Water.

Taken from "Sea Tests of USCGC *Unimak*. Part I—General Outline of Tests and Test Results," by N. H. Jasper and J. T. Birmingham, David Taylor Model Basin Report No. 976, March 1956.

More often the motion is neither so photogenic nor so flamboyant, but equally serious. When a ship is proceeding into a more or less regular train of waves (excessive wave height is not necessary), it periodically attains the proper phase relationship between the downward motion of its bow in pitching and the crest of an oncoming wave with the result that a violent impact is produced between structure and water. The wave neither breaks over the structure nor is the structure lifted clear of and returned to the water. Such action is termed *slamming*. The result is a violent transient vibration superposed on an otherwise gentle oscillation. A typical stress-time record is shown in Figure 5.

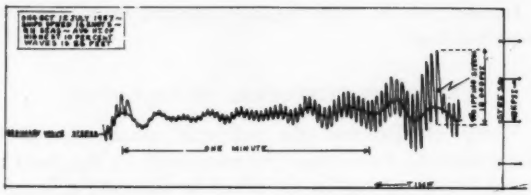


Figure 5. Typical Stress-Time Record on USS *Essex* (CVA9) During Passage Around Cape Horn. Hull Girder Stress Measured at Frame 105½ near Centerline.

Taken from "Strains and Motions of USS *Essex* (CVA9) During Storms near Cape Horn," by N. H. Jasper and J. T. Birmingham, David Taylor Model Basin Report No. 1216, Aug. 1958.

The transient vibrations of ship hulls, or other structural components, resulting from hydrodynamic impulses produced by impact between structure and water surface, as discussed in the foregoing paragraphs, are denoted collectively as *interface reactions, IR*.

A problem somewhat akin to those arising from interface reactions is that colloquially known as the "cable" problem. This area is characterized by transient stresses in the cable connecting two bodies at least one of which is floating at the interface. The floating object may be anchored to the bottom, moored to a buoy, or towing another object, either floating or submerged. What are essential to cause the transient stresses in the cable are the hydrodynamic impulses at one end of the cable. This area will be called *towing-mooring, TM*.

When the problems generated by the fluid interface are considered together with those shown in Figure 3, the state of hydroelasticity at the moment may be considered adequately described.

DEFINITION AND LIMITATION OF HYDROELASTICITY

A casual comparison of Figures 1a and 1b clearly shows how concerted work in a particular field can collect many types of problems not previously considered within its domain. It is equally certain that as ship speeds increase, new configurations are employed, equipment becomes more sensitive to environment, and improved facilities and instrumentation become available to ship research, hydroelasticity will markedly expand its present domain. Surely the dotted portion of Figure 3 gave indication of this.

Following Bisplinghoff's definition of aeroelasticity, *hydroelasticity* is defined as the *phenomena involving interactions among inertial, hydrodynamic, and elastic forces*. Phenomena involving all three types of forces are called *dynamic;* those involving only hydrodynamic and elastic forces are called *static*. Figure 6 represents a "hydroelastic triangle" expected to be valid for at least two decades hence.

Only one sobering note is here injected in order to circumscribe the field. It is easy to extend an

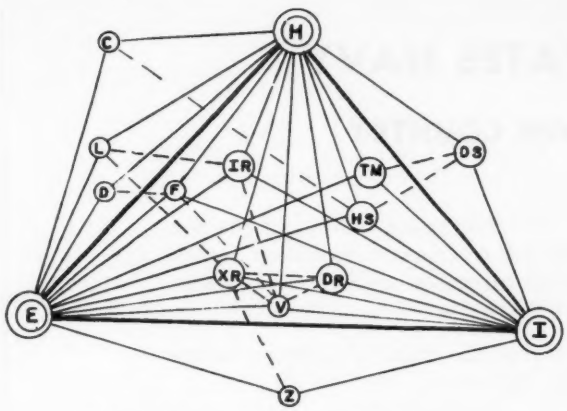


Figure 6. Hydroelasticity Projected to 1980.

interpretation of the definition as stated herein to include virtually all ship structural design. This is specifically and categorically condemned, even though practically all loads considered in the structural design of ships would emanate from one of these three sources. Hydroelasticity must be confined to *mutual interactions*. In short, the necessary and sufficient condition to classify a problem as one of hydroelasticity is that the elastic deformations of the structure induce additional hydrodynamic forces.

CONCLUSION

It seems appropriate to end on a note of challenge. Hydroelasticity, as defined and limited here, is on the one hand an excellent instance of cross-fertilization from several scientific fields, but on the other hand requires a blend of the talents that are peculiar to the naval architect. This is not to say that the hydrodynamicist, structural analyst, physicist, vibration analyst, and the like can be summarily dismissed, but rather that the naval architect with a catholic interest in and understanding of the several areas involved is more highly prized than ever. Hydroelasticity is truly the naval architect's cup of tea.

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An original approach to the design of structural aluminum shapes is contained in an article by H. M. Tiedemann. Working on the assumption that the extrusion process for forming aluminum shapes permits an infinite range of structural shapes, a computer program was evolved to determine the sectional properties of a wide range of possible shapes to permit evaluation. In this manner, hundreds of sections were analyzed in a matter of a few hours of computer time. Preliminary assessment of the results indicates a substantial weight saving with no loss of strength is possible when compared to conventional sections now available.

—from MARINE ENGINEERING/LOG
January, 1959

THE UNITED STATES NAVY

GUARDIAN OF OUR COUNTRY

The United States Navy is responsible for maintaining control of the sea and is a ready force on watch at home and overseas, capable of strong action to preserve the peace or of instant offensive action to win in war.

It is upon the maintenance of this control that our country's glorious future depends; the United States Navy exists to make it so.

WE SERVE WITH HONOR

Tradition, valor, and victory are the Navy's heritage from the past. To these may be added dedication, discipline, and vigilance as the watchwords of the present and the future.

At home or on distant stations we serve with pride, confident in the respect of our country, our shipmates, and our families.

Our responsibilities sober us; our adversities strengthen us.
Service to God and Country is our special privilege. We serve with honor.

THE FUTURE OF THE NAVY

The Navy will always employ new weapons, new techniques, and greater power to protect and defend the United States on the sea, under the sea, and in the air.

Now and in the future, control of the sea gives the United States her greatest advantage for the maintenance of peace and for victory in war.

Mobility, surprise, dispersal, and offensive power are the keynotes of the new Navy. The roots of the Navy lie in a strong belief in the future, in continued dedication to our tasks, and in reflection on our heritage from the past.

Never have our opportunities and our responsibilities been greater.

"THE SHIPBUILDER AND MARINE ENGINE-BUILDER"

TECHNICAL PROGRESS IN MARINE ENGINEERING DURING 1958

ACKNOWLEDGEMENT

This staff article was published in the January 1959 issue of "The Shipbuilder and Marine Engine-Builder."

NUCLEAR POWER

THE PACE of development of nuclear power for marine propulsion is increasing. At the time of the Geneva Conference 1955, the *Nautilus* was the only nuclear-powered vessel at sea, and the nuclear merchant ship was only in the very early planning stage. Only three years later, by the time of the 1958 Geneva Conference, the nuclear submarine had proved itself and, in the United States of America, six more have been launched and seven more are under construction. In the direction of "peaceful uses" the launching has already taken place of the cargo-passenger liner *Savannah* in America and of the ice-breaker *Lenin* in Russia. In Britain and in other countries, valuable papers have been published, closely analyzing the conditions in which a nuclear-powered merchant ship becomes economically attractive to a private shipowner.

Some idea of the size of the first two surface ships to be nuclear powered can be obtained from Table I.

TABLE I.

	Savannah	*Lenin*
Length	587 ft.	440 ft.
Breadth	78 ft.	91 ft.
Displacement, tons	21,800	16,000
S.H.P.	22,000	44,000
Speed, knots	21	18

Normal commercial considerations hardly apply to an icebreaker and it is difficult to imagine a vessel more suited to nuclear propulsion. Existing conventional ice-breakers need frequent bunkering owing to their large powers and high fuel consumption, and their effectiveness is thus greatly curtailed. The *Lenin* will probably have an endurance of three years before refueling is necessary; she is designed to develop a thrust of 330 tons and is expected to be able to cruise at 2 knots in ice 8 ft. thick. The Russian designers consider that, for the sake of reliability, a ship should have two reactors —in the *Lenin* they have actually provided three, one of them being a reserve unit.

The reactors use pressurized water as both coolant and moderator, together with enriched fuel in the shape of uranium dioxide canned in zirconium. The reactor core has a diameter of approximately 3 ft. 3 in., a height of 5 ft. 3 in., and contains 180 lb. of uranium 235. Each reactor generates 90 M.W. (equivalent to 306×10^6 B.T.U. per hour) and boron is included in the center group as a burnable poison, to ease control problems and give a measure of flux flattening.

The *Savannah* will have one pressurized-water reactor, the pressure vessel containing the core being 5 ft. 1 in. in diameter and 5 ft. 6 in. high. The fuel elements consist of sintered uranium oxide containing about 4 per cent of U-235, clad in 0.256 in. stainless steel. Uranium oxide is chosen for the

fuel because it withstands irradiation damage well and is highly resistant to corrosion by hot water so that, even if faults develop in the stainless steel cladding, the attack on uranium oxide will be slow. Because the operation of refueling or changing the core element of the present design of pressurized-water reactor takes a long time, the reactor has been designed for a life equivalent to 600 days at 68 M.W. (heat) generation, *i.e.*, heat rate of 232×10^6 B.T.U. per hour, at the end of which time 48 kilograms of U-235 will have been burned up. For emergency, there are two 750-KW Diesel generators, which are arranged to start automatically upon failure of reactor power; they provide sufficient power for the take-home motor and other essential ship's electrical load. Two 5,500 lb. per hour oil-fired package boilers provide steam at a pressure of 150 p.s.i. for emergency heating requirements.

The principal design performance data for the power plant are given in Table II.

TABLE II.

	Norma.	Maximum Continuous
Shaft power, S.H.P.	20,000	22,000
Shaft speed, r.p.m.	106.7	110
Turbine inlet pressure (dry sat. steam) p.s.i. abs.	480	460
Main condenser pressure, Hg. abs.	1.5	1.5
Feed-water temperature, deg. F.	347	347
Steam generation, total lb. per hour	241,300	259,950
Main turbines, lb. per hour..	186,700	204,625
Other uses, lb. per hour	54,550	55,325
Reactor heat generation, M.W.	63.0	68.3
Reactor pressure, p.s.i. abs....	1,750	1,750
Reactor flow, lb. per hour ...	8,000,000	8,000,000
Primary water temperature rise through reactor, deg. F.	22.6	24.6
Primary water, mean temperature, deg. F.	508	508
Boiler temperature, deg. F....	467	464
Total electrical load, KW	1,850	1,850

The estimated gross weight of the power plant is 3,650 long tons, being made-up from the following: reactor system 600 tons, containment and shielding 1,900 tons, and the propulsion system 1,150 tons.

The reactor, together with its primary loops, pressurizer steam generators, primary circulating pumps, air-conditioning system and other auxiliaries are to be arranged within the main containment vessel. A portion of the reactor system auxiliaries will be just outside this vessel in order to provide easier access for maintenance. The main shielded containment vessel is 50 ft. 6 in. long by 35 ft. diameter, having a cupola, 13 ft. 6 in. diameter, extending upward over the reactor to house the control rod drives and provide access for refueling. The vessel is designed to withstand the pressure of 186 p.s.i. which, it is estimated, could be developed if the entire primary and secondary systems were to rupture. When the reactor is shut down, access is possible through a lower manhole for inspection purposes and through two top hatchways which are large enough to permit replacement of primary pumps and valves.

There is a primary shield in the form of a steel tank, 17 ft. high, such that there is an annular water space of 33 in. width all round the reactor. The secondary shielding is on the outside of the main containment vessel and consists of 30 in. of barytes concrete above the equator of the cylindrical part, the cupola and hatches being provided with 12 in. of lead and polythene. Below the equator the shield consists of 48 in. thick vertical walls of ordinary concrete.

As noted in Table II, the primary loop is pressurized to 1,750 p.s.i. This pressure is maintained by means of an electrically heated pressurizer vessel in which a steam space is maintained in equilibrium with the water by alternate use of electrical heaters or spray coolers, as the system demands. The purification system is designed for the continuous removal of all solids from the primary water, whether active or inactive, and it will also remove fission products that may be present due to defective fuel rods.

When running normally, the reactor system will respond automatically to small variations in steam demand by virtue of its inherent negative temperature coefficient, and no manual control will be necessary. For large changes of demand, such as when maneuvering, the control rods will be used and an automatic system is designed. Automatic shut down will occur due to any one of seven causes; excessive rate of power increase, excessive power, excessive rise or fall in pressure, excessive outlet temperature, loss of flow, loss of power to safety circuits, and loss of power to control rod drives.

Further design details of the *Savannah* are given in a paper by Mr. R. P. Godwin and Mr. W. H. Zinn entitled "The Use of Nuclear Energy for Purposes other than the Generation of Electricity," Geneva Conference P1831, U.S.A., 1958, and in a paper entitled "Safeguard Considerations in the American Nuclear Merchant Ship Program," Geneva Conference, P1023, U.S.A., 1958, by Mr. R. P. Godwin and Mr. D. L. Worf.

The permitted degree of exposure for everyone on board is very low, and in all passenger spaces barely exceeds that due to naturally occurring cosmic and other radiations; it is only one tenth of the permitted maximum rate. Radioactive products collected in the primary filters will be retained on board and will be removed by the bodily removal of the apparatus concerned. The capacity of the filters is expected to be 150 days at full power. Small

amounts of radioactive gases can be discharged through the radar mast.

Details of the operating costs of the *Savannah* have not been published, but in the Andrew Laing Lecture, 1958, Sir John Cockcroft estimated these costs ("Nuclear Power for the Propulsion of Commercial Shipping"; North-East Coast Institution of Engineers and Shipbuilders). American operators can hire enriched uranium fuel from the U.S.A.E.C. at an interest rate of 4 per cent and they also pay for the U-235 burned in the course of the fuel life. The spent fuel elements are returned for chemical processing and refabrication. The operator will be credited with the value of the plutonium in the spent fuel, about 12 dollars per gram. The final estimate is a total fuel cost to the operator of 0.57 pence per S.H.P. hour. Using this figure, a comparison of annual costs for operating conventional and nuclear tankers with a propulsion unit similar to that of the *Savannah* is given in Table III.

TABLE III.

Item	Nuclear, £	Fuel Oil, £
Capital charges on hull and engine	986,000	450,000
Fuel	330,000	200,000
Operating costs	100,000	150,000
Totals	1,466,000	800,000

The figures quoted for fuel oil are typical for a conventional tanker of 47,000 tons deadweight, operating between the Persian Gulf and the United Kingdom. It is assumed that the capital charges are 15 per cent per annum on £3,000,000. On this basis it is seen that £1,000 per annum saving in fuel costs can permit an extra capital cost of £6,700 and still allow the nuclear ship to break even. Thus to be a commercial proposition, a nuclear-powered vessel must be one with a high-load factor (a big tanker, the obvious choice), and the extra capital cost for the nuclear installation must be considerably reduced. The best figures available at present for British practice are given in "Nuclear Propulsion for Ships" by Mr. R. V. Moore and Mr. C. E. Iliffe, Geneva Conference P266 U.K. 1958. The authors discussed the possible applications of the advanced gas-cooled graphite moderated reactor for marine purposes and concluded that fuel costs would be as low as 0.2 pence per S.H.P. hour, but that capital costs of the propulsion unit would be 50 to 100 per cent greater than for conventional propulsion units. Combining these two components they considered that, at 70,000 tons displacement, a nuclear tanker would break even with the conventional tanker.

In the 1958 James Forrest Lecture to the Institution of Civil Engineers, Sir John Cockcroft drew attention to the research study that the Atomic Energy Authority are carrying out on this type of reactor. The aim is to increase the rating of the reactor by using higher gas outlet temperatures, thus reducing capital cost and producing a more compact plant. Fuel element performance is the key to performance and an attempt is being made to develop an all ceramic fuel element to permit the increase in gas temperatures.

Great interest is being shown in the United States of America in the development of the boiling-water reactor, and at 22,000 S.H.P. it is estimated that capital costs would only be 45 per cent greater than for conventional propulsion machinery (100 per cent extra for the *Savannah*). There are, however, several problems in connection with the stability, control and possible contamination of turbine and condenser which have to be answered before such a reactor could be accepted for ship use.

Both the pressurized-water reactor and the boiling-water reactor require enriched uranium fuel and it is not considered possible to produce such fuel in any European country at a cost low enough to enable these types of reactors, even after development to their maximum potentialities, to produce power at an economic cost. Thus, at present, development work in Britain is concentrated on high-temperature gas-cooled reactors.

Detailed development studies have been carried out in Sweden for a nuclear propelled 65,000-ton deadweight tanker of 30,000 S.H.P., using a boiling-water reactor. By using a large core with 24 tons of uranium, a very low grade of enrichment will suffice.

In Japan, design studies have been carried out in connection with an emigrant ship trading between Japan and South and Central America. The use of fuel oil involves 10 per cent extra mileage in order to pick up fuel as there are no bunkering ports on the direct route. A pressurized-water reactor of 180 M.W. output is envisaged and the economics have been based on the assumption that the necessary enriched fuel can be hired from the United States of America on the same terms that are available to United States owners. The possibility of using the same type and size of reactor in submarine tankers has also been the subject of a design study.

Various North German shipyards, shipowners, and industrial undertakings have formed an association for the application of nuclear energy to ships. After extensive study of various models of reactors, the association have chosen a swimming-pool reactor for research purposes on account of its high neutron flux and versatility. The ease with which its internal parts may be inspected and the mobility of the reactor core afford many possibilities for teaching and research purposes. The reactor is designed for a maximum output of 5 M.W., it has 32 fuel elements each containing 0.35 lb. of U-235 in the form of a 20 per cent enriched $U.AL_4$ alloy in between aluminum plates.

The future program of the association provides for the construction, in the next few years, of an

experimental reactor to be run on land under conditions simulating those on board ship. It is then planned to install an actual ship reactor for the propulsion of an experimental vessel.

A masterly summary of the position, and the relative importance for marine work, of various types of nuclear reactors is given in the First De Laval Memorial Lecture given by Dr. T. W. F. Brown in Stockholm in 1957; it is entitled "Propulsion of Ships by Steam Turbine Machinery," and can be read in English in the journal of the Joint Panel on Nuclear Marine Propulsion, Vol. 2, No. 2.

A most useful information bibliography has been published by the U.K.A.E.A. Industrial Group, Risley (Report No. IGRL-IB/R-24, 1958) entitled "Nuclear Propulsion for Ships—Information Bibliography." The interests of both ship and reactor designers have been considered.

GAS TURBINES

An agreement has been reached between the United States Atomic Energy Commission and Maritime Commission and the General Dynamics Corporation for the development of a prototype gas-cooled reactor in combination with a closed-cycle gas turbine plant for merchant ship propulsion. It is probable that the plant will incorporate a high-temperature reactor cooled by carbon dioxide at a pressure of 2,000 p.s.i., the gas being heated from 780°F to 1,300°F.

Carbon dioxide has good heat transport properties and has convenient physical properties for the design of turbines and compressors. It is hoped that such a system will have a thermal efficiency of 34 per cent, whereas the steam turbines used in conjunction with a pressurized-water reactor have thermal efficiencies of only 20 to 25 per cent. The overall weight of the pressurized-water system is also greater.

The reactor is designed for a heat rate of 55 M.W. The fuel material will be a cermet of uranium oxide dispersed in stainless steel. The moderator will be hydrogen in the form of metallic zirconium hydride, a material which has physical properties resembling zirconium, and advantage can thus be taken of the excellent moderating properties of hydrogen at temperatures where it would be impossible to contain liquid water. The Atomic Energy Commission estimate that it will require from five to seven years to make the necessary advances in reactor technology to construct an actual installation.

The performance of a closed-cycle gas turbine such as is proposed herewith will greatly depend on the performance of the regenerator heat exchanger, as quite small pressure losses can easily eliminate the thermodynamic gains due to its inclusion. Size too is a major consideration and the manner in which pressure drop and bulk of regenerator is affected by the choice of working fluid is the subject of an article by Mr. P. Fortescue and Mr. D. V. Wordsworth in *Engineering*, Vol. 185 (1958), 28th February on "Gas-turbine Regenerator Performance."

More information about the new *Auris* machinery has been published in the American Society of Mechanical Engineers' Paper No. 58-GTP-12 of the Gas Turbine Power Conference, March 1958. The paper is entitled "The *Auris* Gas-turbine Project," by the late Mr. J. Lamb and M. L. Birts. The first part of the paper gives a detailed summary of the five years experience with the first gas turbine set installed in the *Auris*, and the second part deals with the new 5,500-H.P. gas turbine set which is to be the sole means of propelling the ship.

The new turbine, designed by the British Thomson-Houston Co., Ltd., and built by them and by Messrs. Cammell Laird & Co. (Shipbuilders and Engineers), Ltd., is a development of the earlier turbine and is similar to it in many fundamental features. It is an open-cycle installation, designed for long life, burning any grade of residual fuel. It has multistage turbines with uncooled rotors and casings, the combustion chamber is refractory lined and the turbine inlet temperature is unaltered at 1,200°F. Intercooling has been adopted and the heat exchanger has been relatively increased in size. After passage through the heat exchanger, the exhaust gases pass through a waste-heat boiler, the steam from which drives a 200-H.P. turbo generator. Some of the features are given in Table IV.

TABLE IV.

Maximum power at turbine coupling, B.H.P.....	5,500
Maximum power available for propeller, S.H.P...	5,300
Thermal efficiency based on B.H.P. at turbine coupling, per cent	26.3
Thermal efficiency as above but including 200 H.P. for turbo generator, per cent	27.2
Thermal efficiency of the previous set per cent...	22.1

The conclusion of the authors included the following. A gas turbine for merchant ship propulsion must not be selective in the fuel it uses, and this factor plays a large part in determining the most suitable configuration and construction of the turbine. The difficulty of avoiding deposition on, and corrosion of, turbine blading becomes much greater at gas temperatures over 1,250°F and, as no simple and universal solution has yet been found, a maximum temperature below this figure is strongly indicated. To compensate for the limited cycle efficiency, the component efficiency of the turbine must be as high as possible, and, as insensitivity to deposition requires wide gas passages and consequently low-deflection blading, the turbine must be of the multi-stage type. The discovery of a fuel additive which would prevent deposition and corrosion at temperatures above 1,250°F would modify this argument and would still further improve the prospect for the marine gas turbine.

An interesting and up-to-date summary paper appears in the Transactions of the Institute of Ma-

rine Engineers, Vol. 60, No. 10, October, 1958, entitled "Gas Turbines for Merchant Ship Propulsion," by Mr. S. Kasthuri. Among the plants discussed are those on the *Auris, John Sergeant,* the Pametrada design and the Sulzer design.

One of the new Ruston Mark TE, 430 B.H.P. gas turbines is installed for auxiliary electric power generation on the motorship *Weybridge,* built by Messrs. Barclay, Curle & Co., Ltd., for Messrs. Watts, Watts & Co., Ltd. It works on a non-recuperative simple open-cycle and has been designed for long life, about 25,000 hours full load operation being expected before complete overhaul. Details of the unit are given in Table V.

TABLE V.

Maximum inlet temperature, deg. F.	1,470
Pressure ratio	3.5:1
Thermal efficiency (on B.H.P.), per cent ..	12.0
Gear box reduction, r.p.m.	19,250/1,200 or 1,500

The problems of thermal expansion and insulation are accentuated in gas turbines as operating temperatures are increased. A paper by Mr. L. T. Whitehead, entitled "The Mechanical Design of an Experimental Plant-type Gas Turbine" is published in the Transactions of the Institution of Mechanical Engineers. The paper emphasizes the features embodied in the design and construction of an 8,000-B.H.P. gas turbine which are aimed at overcoming these problems. The work is being carried out at the National Gas Turbine Establishment. Rotor discs are connected by threaded side rims of large diameter and cooled by an axial-flow of air under the blade roots. Flexible sleeves are fitted between the shaft journals and the roller type bearings, to inhibit heat flow and to accommodate differential radial expansion. The stator casings are treated as gas carrying ducts and are relieved of mounting constraints and bearing alignment duty by being freely mounted on radially-disposed pegs and keys allowing freedom for thermal expansion in all directions. Preliminary experiments on the bearing mounting sleeves, and on the surface treatment to prevent seizure of the screw-thread connections between rotor discs, are described. Great attention is given in the design to methods of facilitating inspection and maintenance and provision is made for replacing individual rotor blades and bearings without separating the rotor and stator.

STEAM TURBINES AND BOILERS

An unusual turbine set is installed in the bulk-cargo steamship *Hugo Stinnes.* The set has been designed and built by Maschinenfabrik Augsburg-Nürnberg A.G.; turbines, condensers and gearing form a single, rigid, flexibly-mounted block; high pressures and high shaft speeds are used. The two main boilers are normally rated at just over 40,000 lb. of steam per hour at 1,225 p.s.i. and 950°F. The three-casing set, with double-reduction gearing, develops 9,000 S.H.P. at 115 r.p.m. and 10,000 S.H.P. at 119 r.p.m. Details of the set are given in Table VI.

TABLE VI.

Turbine	Type	Steam, p.s.i.	Conditions, deg. F	H.P.	R.P.M.
H.-P. ..	4-stage impulse	1,120	930	2,500	14,300
I.-P. ...	4-stage impulse	—	—	2,800	12,500
L.-P. ..	8-stage impulse	125	500	5,100	4,000

The gearing has been supplied by Zahnräderfabrik Renk, of Augburg; it is of articulated type, both h.-p. and i.-p. primary pinions engaging with the same primary wheel. The total weight of the gearing is 87 tons.

A new design of turbine set, running at more orthodox speeds has been developed by Allgemeine Elektricitats-Ges. The S.H.P. is in the 16,000 to 17,000 range at 100 to 103 r.p.m., and the set is designed to take steam at 835 p.s.i. and 850°F. The h.p. and l.p. turbines run at 5,500 r.p.m. and 3,800 r.p.m., respectively. The astern turbine is arranged within the exhaust outlet of the l.p. turbine; its wheels are mounted on keyed bushes and secured with radial dowels. Such a construction permits the astern turbine to be started up rapidly, even when not fully warmed through, and it can develop 75 per cent of ahead torque at 65 r.p.m. The non-bleed steam rate is quoted as 16,0000 S.H.P. at 6 lb. per S.H.P. hour.

A valuable contribution to the published material concerning design of a tanker turbine installation has been made by Commander A. D. Bonny, R.N. (Rtd.), in his paper presented to the Institute of Marine Engineers in October, 1958, entitled "An Investigation into the Optimum Machinery Installation for a Large Steam Turbine Tanker." The object of the investigation is to find the installation which develops 15,000 S.H.P. and which best meets the requirements of simplicity and low cost, low maintenance and low fuel rate.

As a basis of comparison, a tanker of typical power with a simple feed system (600 p.s.i. and 850°F.) is chosen and a possible operating schedule set out. The optimum pressure for a given steam temperature is investigated and the effects of costs and maintenance examined. Full account is taken of the auxiliary loads together with their margins. The use of regenerative feed and air heating are considered and the effect on the basic cycle of over 15 items is listed. These alternatives are then added in various ways to give 10 different feed cycles with an estimated first cost and estimated saving in the annual fuel bill.

From this an owner can make a final choice of feed cycle and there follows a discussion of principal components and certain aspects of water purity, automatic control and standardization.

It is concluded that by increasing the tempera-

tures to 900°F and cutting unnecessary margins, it is possible to produce a design cheaper, and with an oil rate 2,000 tons per annum less, than the basic. The achieved specific fuel consumption is estimated to be 0.525 lb. per S.H.P. hour, although with a little extra cost this could be further improved by using a steam temperature of 950°F.

The American Society of Mechanical Engineers' paper No. 57-A-264, December, 1957, by Mr. A. O. White, and entitled "The Combined Gas Turbine—Steam Turbine Cycle with Supercharged Boiler and its Fuels," refers primarily to land power plant, but it is equally suggestive for marine application. The gas-turbine driven compressor delivers to the boiler the whole of the air required for fuel combustion at a pressure of four to six atmospheres. At boiler exit, the products of combustion, still at pressure and at a temperature up to 1,450°F, pass on to the gas turbine through which they expand to atmospheric pressure, the heat remaining in them being re-cycled either through air or feed-water heaters. The gas turbine drives the compressor and a generator. The estimated improvement in cycle efficiency is by 4 to 8 per cent and for the same steam flow to the steam turbine, the net plant output is increased by 15 per cent. Of course, forced and induced-draft fans are eliminated. The component which will differ most from conventional units and has to be specially designed is the supercharged boiler, but most major boiler manufacturers have investigated the problems involved and have made preliminary designs for suitable units.

Such a cycle offers considerable size and weight saving over a conventional steam cycle. In the American Society of Mechanical Engineers' paper No. 58-SA-25, June, 1958, entitled "Design and Development of a Supercharger for a Pressure-fired Boiler," the authors present the mechanical and thermodynamic design features of an axial-flow supercharger for a pressure-fired boiler capable of generating 132,000 lb. of steam per hour at 1,200 p.s.i.g. and 1,000°F.

The eleven-stage axial-flow compressor is designed to supply 40,000 cu. ft. per minute at 4.5 atmospheres, and to operate over a wide speed range. The power required is 6,000 H.P. and is furnished by an integrally-cased two-stage gas turbine taking the boiler outlet gas at 61.2 p.s.i.a. and 815°F. The shop tests of the supercharger are given.

The reduction in space and weight is largely due to the fact that the gas side of the heat transfer system is under high pressure, resulting in increased heat transfer rates in the radiant heat absorbing part of the furnace, as well as the convection part of the boiler. A paper entitled "Supercharged Boilers for Marine Application," presented by Mr. E. L. Daman to the Institute of Marine Engineers in April, 1958, deals with the system in which the gas turbine and compressor is just self sustaining so that there is no call for external power supply for the combustion air supply. The design and performance characteristics of the supercharged boiler are discussed in the paper and operational results are quoted from an experimental set with a capacity of 125,000 lb. per hour at 1,200 p.s.i. and 950°F, which had been operating for over one year (last April). The results compared favorably with those from a conventional plant for the same duty, over the whole range of 10 to 120 per cent full load. A comparison of weight and space requirement is given in Table VII.

TABLE VII

		Supercharged	Conventional
125,000 lb. per hour	Dry weight	61,500 lb.	116,250 lb.
1,200 p.s.i.	Wet weight	67,900 lb.	126,500 lb.
at 950° F	Box volume	2,600 cu. ft.	5,280 cu. ft.
52,500 lb. per hour	Dry weight	35,000 lb.	216,500 lb.
600 p.s.i.	Wet weight	40,500 lb.	231,100 lb.
at 850° F	Box volume	1,870 cu. ft.	8,030 cu. ft.

Tube withdrawal spaces are allowed for, but the blowers have been omitted for the conventional figures. The experimental unit had been run on distillate fuels only, but the author reported an investigation, using a small test boiler, into the possibility of burning heavy residual oils in supercharged boilers. The tests show very promising results in the use of magnesium-aluminum additives to prevent heavy accumulation of ash on the screen and superheat tubing.

The problem of matching a supercharger set to a supercharged boiler is discussed, and numerical data are quoted to show the effect of mis-matching on boiler weight and performance. Some important requirements of the control system are mentioned. The general conclusion derived from experience is that supercharged installations have features making them attractive for use on merchant ships, and that reliability can be achieved, comparable with that of conventional generating equipment.

Considerable work has been carried out on the British Shipbuilding Research Association's experimental two-tube water-tube boiler, built to investigate natural circulation phenomena and designed for a maximum pressure of 1,500 p.s.i., and a maximum heat input rate of 120,000 B.T.U. per hour sq. ft. Some of this work is reported in the paper presented to the North-East Coast Institution of Engineers and Shipbuilders in March, 1958, by Mr. A. M. Laird, Mr. A. W. Scott and Mr. A. S. T. Thomson, entitled "Natural Circulation Investigations on an Experimental Two-tube Boiler." The main series of experiments shows the influence of tube diameter, pressure and heat-input rate on circulation velocities, and additional tests indicate the effects of restricting orifices in the circuit, non-uniform heating of the riser, shortening the heated length and rapid drop in boiler pressure.

The following general results were reported:

Under steady operation, natural circulation was more than adequate to deal with the maximum heat loading and evaporation rate attained in the tests.

Tube diameter had an important influence on circulation rate at all conditions studied.

For the 2¼-in., 1¾-in. and 1¼-in. tubes the maximum circulation velocities attained were approximately 5 ft., 4 ft. and 3 ft. per second.

At pressures above 400 p.s.i. and heat input rates in excess of 40,000 B.T.U./hour sq. ft., changes in pressure and heat input rate became relatively less important.

Non uniformity of heating around the riser tube was unimportant.

The rate of drop in pressure to produce circulation failure was related to boiler pressure and to the circulation velocity obtaining before the pressure drop occurs.

"Materials for Superheater Tubes and Supports" is the title of a paper by Mr. D. W. Crancher, read before the Institute of Marine Engineers in January, 1958. The trend to higher steam temperatures and the deterioration in the quality of fuel oil has made the problem of the choice of suitable materials for superheater tubes and supports one of the most pressing boiler design problems. The properties required are creep and corrosion resistance in both steam and flue-gas atmospheres; adequate mechanical strength at room temperature; ease of welding; adequate strength of welded joints; and availability. Six tube steels have been developed in this country, and the author discusses their properties. The steels are:

(1) Plain carbon steels (B.S. 494 : 1950, B.S. 512 : 1950, B.S. 1654 : 1950).

(2) 0.5 per cent molybdenum steel (B.S. 1652 : 1950).

(3) 1.0 per cent chromium, 0.5 per cent molybdenum steel (B.S. 1653 : 1950).

(4) 2¼ per cent chromium, 1.0 per cent molybdenum steel.

(5) 0.5 per cent molybdenum, 0.25 per cent vanadium steel.

(6) 18-21-1 (chromium/nickel/niobium) austenitic steel.

No known commercial metal is immune from corrosion due to attack by molten fuel ash, and the choice of materials for supports (as distinct from tubes) is greatly complicated by the effect of ash attack. In the absence of such attack, heat resisting steels are available with adequate strengths up to 1,800°F. The author discusses methods of combating ash attack (and the associated problem of tube fouling) and suggests that the most promising solution is to treat the fuel in order to modify the ash formation. Two such treatments are referred to, in particular (1) the desalting system, which involves washing with magnesium sulphate solution followed by double centrifuging, and (2) various additives, whose purpose is to raise the melting point of the ash and, at the same time, to tie up the sodium and vanadium in forms which do not form deposits or corrosion; the ideal one has not yet been found.

With advancing conditions of steam generation, the elimination of all sources of contamination of feed-water becomes of utmost importance, and in an article in the *Syren and Shipping* of 30th April, 1958, on "Care and Maintenance of the Water-side of Marine High-pressure Boilers," the author draws attention to sources of contamination which are often missed. The need for thorough pre-commissioning cleaning of all units, piping and tanks is stressed, as is also the need to provide initial temporary protection against corrosion of such items before they are installed. It is recommended that prior to steaming for the first time, the boilers should be either acid cleaned, or degreased and all atmospheric oxidation removed. It is assumed, of course, that a proper water treatment is used for normal boiler operation.

The Test Code for Acceptance Tests for Steam Turbines—B.S. 752: 1958—has been published. It is a standard which enables the guarantees given by makers of steam turbines in accordance with B.S. 132 (steam turbines) to be verified. Test of output, steam consumption, speed regulation and emergency governor operation are specified. Notes are appended describing suitable instruments and methods of measurement, *viz.*, dynamometers, electrical instruments, weighing tanks, volumetric measuring tanks, surface condenser leakage tests, and allowances for other leakages.

FREE-PISTON GAS-TURBINE MACHINERY

The first complete free-piston gas-turbine propulsion plant to be installed in Great Britain has now undergone shop trials, and is fitted on board the ore-carrier *Morar*, a vessel 407 ft. in length and of 9,200 tons deadweight. The service speed is expected to be 11 knots. The *Morar* thus becomes the first ocean-going merchant ship to be powered by free-piston machinery from the commencement of its service. The propelling plant comprises three GS-34 gas generators, built by the Free Piston Engine Co., Ltd., designed to yield 1,230 G.H.P. at full load; one gas turbine (developing 2,500 S.H.P. at 5,250 r.p.m.), built by Messrs. Rankin & Blackmore, Ltd., to the design, and with the technical assistance of Power Jets (Research and Devolpment) Ltd.; and a set of double-reduction gears of articulated locked-train type, built by the Fairfield Shipbuilding & Engineering Co., Ltd.

Of new design, the gas turbine is a simple two-bearing unit with ahead and astern turbines on the same shaft having separate inlets and a common exhaust. It is intended to use fuel of 1,500 secs. Redwood No. 1 at 100°F in service, and even heavier oils. The estimated weight of the propulsion machinery installation is 410 tons, as compared with a corresponding estimate of 630 tons for a Diesel-engined sister ship.

Considerable experience is being gained in operating gasifiers on heavy fuel oil. The *William Patterson* is reported to be using Esso Bunker C with-

out limitation on ash or sulphur content. On the trials the figures quoted were 0.95 per cent ash, 2.48 per cent sulphur, 3,600 secs. Redwood No. 1 at 100°F, and 0.9574 specific gravity. No special treatment is provided other than heating, and passing through one of two Sharples D.H.2 centrifuges with a capacity of 480 gallons per hour. The highest recorded power during a voyage is 5,900 S.H.P.

When running on heavy fuel up to 3,000 secs. Redwood No. 1, at 100°F, measurements on the exhaust side of a GS-34 gasifier have shown a wear rate of 0.01 in. per 1,000 hours, or less, on the first piston ring of the engine, and a wear rate of 0.003 in. for the cylinder liner. The continuing favorable results obtained from existing installations, both shore-based and marine, indicate that the reliability of the free-piston gas generator is comparable with that of the Diesel engine, and, of course, accessibility and ease of maintenance is superior. Considerable information about existing sets of free-piston machinery is given in the following two papers:

"Free-piston Progress," by Mr. F. A. I. Muntz and Mr. M. Barthalon, before the Institution of Engineers and Shipbuilders in Scotland, December, 1957.

"Present State and Future Outlook of the Free-piston Engine," by Mr. R. Huber (American Society of Mechanical Engineers' paper No. 58-GTP-9, at the Gas Turbine Conference, Washington, March, 1958).

Both papers examine the trend of future development, particularly in regard to increasing the power output. This can be achieved by supercharging the standard GS-34 with a turbo blower, by after-burning between gasifier exhaust and turbine inlet, or by the development of altogether larger units. Mr. R. Huber gives a general formula relating the dimensions and principal characteristics for engines similar to the GS-34 and then discusses a gasifier with a 20-in. Diesel bore. Such an engine, supercharged, twinned and with after-burning, would produce 8,000 S.H.P. on the turbine shaft.

An altogether different way of obtaining higher power is the use of a plant working on the mixed gas/steam cycle. This would involve a group of gasifiers supplying a steam turbine and making use of the gas (which is 80 per cent unburned air) as the highly preheated combustion air in which more fuel is burned for steam generation. In this way a power output about four times that given by the gas generators only would be obtained.

An interesting article on certain theoretical aspects of the free-piston gas generator appeared in *Engineering*, Vol. 184, of the 29th November, 1957; the author is Mr. J. R. Gingham. In the first section the calculation of performance in terms of the main design variable, *viz.*, compressor pressure ratio, is discussed and various approximations are made for simplicity. The second section describes the procedure for estimating the main dimensions of a gas generator of specified output. Then follows an outline of the principles by which the power output of an existing design can be modified by scaling the design up or down or altering the piston bulk density. Finally, the author discusses the procedure for estimating performance at part load or other conditions away from the design point.

The Fiat Company are designing and experimenting with a free-piston gas generator capable of developing 2,000 S.H.P. on the turbine shaft.

"Marine Gas Turbine and Free-piston Turbine Bibliography," by Mr. J. W. Sawyer in the JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS (February, 1958) provides brief summaries of 108 papers. These have been arranged by years, from 1952 to 1957 inclusive. A subject and author index is included which covers about 450 subjects and provides over 1,050 references.

DIESEL ENGINES

The first Clark-Sulzer turbo-charged Diesel engine has been fitted on board the *Blyth Adventurer*. This engine, manufactured by Messrs. George Clark and North Eastern Marine (Sunderland), Ltd., is the first British turbo-charged engine of its type (RSAD76). A seven-cylinder engine, it develops a maximum continuous output of 9,100 S.H.P. at 119 r.p.m., the normal service output is to be 8,225 S.H.P. at 115 r.p.m. The bore and stroke are 760 mm. and 1,500 mm. It is fitted with two Brown-Boveri turbo blowers (type VTR.630) and is equipped to burn boiler oil of up to 3,500 secs. Redwood No. 1 at 100°F.

The second Doxford pressure-charged engine has been commissioned and is installed in the 19,000-ton tanker *Spinanger*. With a b.m.e.p. of 100 p.s.i., a service output of 8,000 S.H.P. is developed at 115 R.P.M. by this six-cylinder engine; a mechanical efficiency of 91.5 per cent is claimed. The bore is 650 mm. and the combined stroke 2,320 mm.

In October, 1958, Dott. Ing. Roberto De Pieri presented his paper on "Recent Developments in Italian Marine Diesel Engines," to the Institute of Marine Engineers. The design characteristics of the Fiat Company's most modern engine—a 14,400-S.H.P., single-acting, 12-cylinder, turbo-charged unit—are discussed. The bore is 750 mm. and the stroke is 1,320 mm. Supercharging is by means of constant-pressure turbines driving blowers which work in series with reciprocating air pumps. Three Fiat-Brown Boveri turbo-chargers (type VTR.630) are used on the 12-cylinder engine and are arranged so that the accessibility of the cylinder heads is the same as for the unsupercharged engine. The air is cooled after the turbo compressor and also after the reciprocating air pumps. Test results are quoted in the paper; an indication of the performance is given in Table VIII.

TABLE VIII

	Total S.H.P.	S.H.P./ Cylinder	M.E.P., p.s.i.	Exhaust Temperature, deg. F
Normal rating	14,400	1,200	100	644
Overload	17,000	1,400	112	716

In common with other builders of large Diesel engines, the Fiat Company are in the process of developing an engine capable of delivering about 2,000 S.H.P. per cylinder. The greatest difficulties will probably be of a thermal nature, developing from the larger dimensions of the parts subjected to high temperatures. Experiments are being carried out on a new method of piston cooling and the construction of an experimental unit with two cylinders of 900 mm. diameter and 1,600 mm. stroke is expected to be completed in 1959. It is hoped to develop 1,900 H.P. per cylinder at 118 r.p.m., that is with an m.e.p. of 104 p.s.i. and a mean piston speed of 1,240 ft. per minute. This engine type will be built only for supercharging. It is reported that a nine-cylinder engine, developing 1,700 S.H.P. per cylinder, normal rating, has been ordered by an Italian shipping company, and will be installed in a 36,000-ton tanker.

At least 18 sets of M.A.N. highly-supercharged engines have been installed in ships; the first one has been in operation for 2½ years in the motor vessel *Lichtenfels*. These are, of course, medium-speed, four-stroke cycle engines with a full load output of 2,800 S.H.P. at 250 r.p.m.; the m.e.p. is 227 p.s.i. and the fuel consumption is 0.31 lb. per S.H.P. hour. The maximum pressure is 1,210 p.s.i. At present the highly supercharged two-stroke cycle engine is mainly in the experimental stage. It is clear that supercharging up to admission pressures of 4 to 5 p.s.i. can be effected with little change in design of cylinder, pistons, running gear and crankshaft, but if the admission pressure is raised from 10 to 14 p.s.i., then all of these items have to be redesigned as the maxium pressure goes up, and also the thermal loads.

The matching of a turbo charger and a two-stroke cycle Diesel engine is a complex problem requiring lengthy calculation, since there must be a state of equilibrium between engine, turbine and compressor for each operating point. In an article by Mr. A. A. Reinhart in the *Motorentectnische Zeitschrift* (Stuttgart) of January, 1958, entitled "Rapid Approximate Method for Matching Turbo Chargers to Two-stroke Diesel Engines," the solution to this problem is approached by means of three nomograms, which are based on relations derived from energy and flow balances of engine, turbine and compressor. Certain simplifying assumptions are made and are clearly indicated. The procedure was first developed in connection with extensive charging trials on fast-running, uniflow scavenged, two-stroke cycle engines, and was later applied successfully to cross-flow scavenged two-stroke cycle engines of medium and large size. A worked example is given.

A combined flame trap and crankcase relief valve, developed by the British Internal Combustion Engine Research Association, is now being produced commercially by the Pyropress Engineering Co., Ltd., of Bedford (a B.I.C.E.R.A. partition flame trap, used to separate individual crank chambers is also made by the same firm). The relief valve is offered in three sizes; the valve body projects into the interior of the crankcase and to fit it, it is only necessary to cut a circular hole in the crankcase door.

The principle of operation of the valve is as follows. In the event of the pressure within the crankcase rising to not more than 2 p.s.i., the valve opens and discharges the gases to atmosphere *via* (*a*) six layers of oil-wetted gauze, (*b*) the opening of a spring-loaded circular member, with an air seal around its periphery and an oil seal around its guide on the spindle, and (*c*) a deflector cowl which directs the gas downwards and not across the gangway beside the engine. The effect is to quench an explosion flame and dissipate the force harmlessly. As pressure within the crankcase falls, the coil spring returns the circular member to its closed position.

MISCELLANEOUS

Corrosion

Paint will remain the chief means of protection for all parts of a vessel above water, but for the submerged part of the hull it is probable that cathodic protection will be used as a supplement to the protection afforded by anti-corrosive bottom compositions.

Despite numerous publications on this subject, there still remains a diversity of opinion concerning the criteria for cathodic protection. A laboratory investigation carried out for the American Bureau of Standards gave major significance to potential. Optimum protection was achieved when specimens were controlled at —0.77 volt with reference to the saturated calomel half cell.

The British India Steam Navigation Company's steamship *Kenya* has been fitted with the F. A. Hughes' Guardion external hull scheme for over two years and the results achieved can now be assessed. Sixteen stream-lined groups of magnesium anodes are fitted; seven groups on each side at the turn of the bilge and an additional group on each side above the rudder post. When originally fitted, the *Kenya* was badly scaled, with heavy wastage of rivets around the bows. The results have been excellent; old scale removed, pitting suppressed, and the painted surface of previously descaled plates maintained in excellent condition.

The original anodes were renewed in February, 1958, after 18 months' service. Only two of the holding studs had been damaged and needed replacement, otherwise the new anodes were bolted direct to the original studs. The normal painting procedure has been able to be reduced. At the last two dockings bare metal patches have been given one coat of underseal and the hull has then been coated with anti-fouling, dispensing with the normal coat of anti-corrosive paint. Nearly 200 ocean-going ships have been fitted with such external hull protection.

The *Kenya* is an example of a sacrificial self-regulating method of protection.

Auxiliaries

The motor vessel *Hindustan* is one of the first, if not the first, British-owned dry-cargo ships wholly equipped with a three-phase installation, and in which squirrel-cage motors have been used exclusively. In all, about 70 three-phase squirrel-cage motors are on the ship, with outputs ranging up to 52 KW; all are started by means of circuit breakers or push buttons. Current is supplied by three constant-voltage alternators, each with an output of 250 KVA. at 400 volts. The excitation is dependent on the load, so that voltage fluctuations, even when starting a number of winches simultaneously, is rapidly smoothed out and they have no detrimental effect. Direct current for excitation is supplied by a rectifier.

It would seem that the arrangements for auxiliary power on the Diesel-driven tanker *Butmah* are likely to become standard for high-powered Diesel vessels. The exhaust gas boiler of the 11,250-S.H.P. supercharged engines generates enough steam to run a 400 KVA turbo alternator. A similar sized Diesel alternator is installed for port use, but it is more economical to run the turbo alternator at sea.

Atlas Maskinenfabrik, of Copenhagen, are now building a steam turbine for auxiliary power generation in four standard sizes, covering the range 50 to 1,500 KW.

Cargo-handling

One of the most striking results due to improved cargo-handling technique has been provided by the vessels *Ofelia, Celia* and *Valeria* which trade between Gothenburg, Leith, Newcastle-on-Tyne and Liverpool. They are about 200 ft. long with three hatches of similar size, closed on the weather deck by covers of orthodox design; the 'tween deck hatches are closed by five panel-folding, steel, flush covers. Cargo is handled by three 3-ton electro-hydraulic cranes, each with an outreach of 50 ft. The vessels have been designed to obtain the maximum advantage from handling fully palletized cargo.

The overall result is that, on the run Gothenburg to Leith and Newcastle-on-Tyne, the duration of a round trip has been reduced from 14 days to seven days. A considerable outlay in pallets has been necessary, but the great increase in rapidity of turn around makes it worth while.

A tandem-compound, 3600 rpm, turbine now under construction is believed to be the largest in the world. It will have a rating of 325 megawatts, driving a generator of 384 mva. Planned for installation at Helena, Arkansas, the turbine will be provided with steam at 2000 psig and 1000°F. It will have reheat to 1000° F, exhaust to 1½ inches Hg absolute and will consume over two million pounds of steam per hour.

—from WESTINGHOUSE ENGINEER
January, 1959

A technique of plastic-coating motor winding wire and potting the joints with epoxy resin has enabled the Westinghouse Corporation to develop a completely "wet" electric motor. This insulation can maintain a resistance of hundreds of megohms even when submerged under pressure. Thrust and guide bearings are designed for water lubrication, the thrust bearing taking loads up to 4500 psi. The water environment serves as an effective heat transfer medium. Present sizes are 40 to 75 hp, but motors up to 450 hp are contemplated.

—from WESTINGHOUSE ENGINEER
January, 1959

"THE MOTOR SHIP"

MARINE DIESEL ENGINEERING IN 1958

ACKNOWLEDGEMENT

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THE GENERAL lull which has extended over 1958 in the ordering of new ships has at least given the shipping, shipbuilding and marine industries an opportunity to pause for reflection, for re-appraisal on experience gained and the effect on subsequent ships. For this has been a year when economy of ship operation has once again been the deciding factor as to whether or not a ship can remain in a competitive market. The saving of a few "negligible" tons of fuel oil daily, the extra tons of weight of the main engine, the additional cylinders incurred by avoiding the adoption of pressure-charged engines, the inefficient drive of cargo pumps and many other items are now combining to decide whether the vessel makes a profit or a loss.

No essentially new designs of marine Diesel engines have been introduced over the past 12 months which have, however, seen a much greater demand for units of high powers. Service outputs of 15,000 b.h.p. are no longer uncommon, and this year will see the first large-bore marine engines running which ultimately will give 20,000-22,000 b.h.p. Orders for non-turbocharged engines are dwindling and doubtless the next few years will see the almost complete extinction of the normally aspirated unit. The reliability of the turbocharged unit has now been established; it is becoming inherently simpler to build and maintain, it is more compact, generally has a lower specific fuel consumption and in all, offers greater scope to the naval architect. Thus, there remains little to warrant the retention of the non-turbocharged marine propulsion engine.

The saying that demand creates supply is particularly apt in the case of the modern two-stroke turbocharged engine which came into being as the result of the persistent requirements of shipowners, mainly Scandinavian, for Diesel engines of higher powers. With characteristic independence, such owners showed disinclination to follow the example of British oil companies and those of U.S.-owned or controlled concerns, generally without Diesel engineering experience, who specify turbine machinery and who only now are showing tendencies to reverse the established custom and follow the lead of the independent tanker operators.

MOTOR TANKER OF HIGHER POWERS

Despite earlier and repeated contentions that the inherent steam requirements of a tanker favored the steam turbine, such convictions have been weakened considerably by the unabated interest of tanker operators—the independents, excepting a few noticeable oil concerns in France, Belgium and, more recently, Denmark—for single- and twin-screw Diesel propelling machinery of powers up to 20,000-22,000 b.h.p. Although on first impressions it would have appeared that single-screw machinery must inevitably be the more efficient for the largest classes of tanker, in some quarters it is thought that propeller considerations will curb the trends towards a general adoption of single-screw vessels

of 20,000 b.h.p. to 22,000 b.h.p. and above—whether Diesel or steam propelled. Thus, it may well be that with the present restricted degree of pressure-charging of marine Diesel engines, sufficient power can be developed to meet the powers which can be absorbed efficiently by the propeller or propellers of the average-speed merchant ship. Demands for high output per cylinder persist not only from tanker operators but from owners of cargo liners desiring 10,000 b.h.p. or thereabouts in six cylinders, and a number of such engines are now on order.

At the time of writing, no marine Diesel engine of above 15,000 b.h.p. is in service, although a great number of that power, and higher, are on order. The highest-powered motor tanker delivered in 1958 was the twin-screw 20,000-b.h.p. "Sven Salén," of 40,000 tons d.w.c., and propelled by two Götaverken engines, each of 10,000 b.h.p. at 115 r.p.m. The commissioning of this vessel was, however, of twofold interest, as it also paved the way for the subsequent announcement that one of Britain's largest marine engineering concerns, the North Eastern Marine Engineering Co., Ltd., had concluded an agreement to build Götaverken engines, with powers of up to 22,000 b.h.p. A prototype six-cylinder engine with a bore of 850 mm. and a piston stroke of 1,700 mm. is being built at the Gothenburg works for delivery in 1960, the rated output being 11,000 b.h.p. The Götaverken engine is a longitudinally scavenged unit which, when turbocharged, employs the constant-pressure system and retains the one double-acting pump per cylinder. In the course of the year, two more firms have acquired licences to build Götaverken engines, these being Kieler Howaldtswerke, who are especially interested in the 850-mm. bore engine, and Soc. Anon. des Forges et Chantiers de la Méditerranée.

HIGH-POWERED B. AND W. ENGINES

The B. and W. engine, firmly established in its turbocharged form, has been specified extensively for powers of 15,000 b.h.p., a number of this power being in ships now operating or launched while more than 40 remain on order. These are 12-cylinder units with a bore of 740 mm. and a piston stroke of 1,600 mm. The first ship in the world to be propelled by a 15,000-b.h.p. Diesel engine was the *Yuyo Maru No. 5*, completed in Japan in 1958 and propelled by a Hitachi-B. and W. unit. In August, the highest powered marine Diesel engine built in Europe ran trials at the Copenhagen works of Burmeister and Wain, this having the same cylinder sizes and rated at 15,000 b.h.p. at 115 r.p.m. with a mean effective pressure of 7.1 kg./cm.^{2} (100 p.s.i.), and the mechanical efficiency 89.5 per cent. This has a weight of about 600 tons, or 90 lb. per b.h.p. With the engine developing 16,000 b.h.p. on test, the fuel consumption was not greater than 160 gr. (0.35 lb.) per b.h.p.-hr., the minimum, at 10,000 b.h.p. being 155 gr., or somewhat under 0.345 lb.-hr. This will shortly enter service in the tanker *Kongsvang*, chartered for 10 years by the Belgian oil company Petrofina, which has ordered a 45,000-ton tanker from Burmeister and Wain to be propelled by a 12-cylinder engine with a cylinder bore of 840 mm., a stroke of 1,800 mm. and an output of 1,730 b.h.p. per cylinder in continuous service at 110 r.p.m. with an m.i.p. of 8.0 kg./cm.^{2}

Another such unit, but with 10 cylinders, has been ordered by Sig. Bergesen d.y. and Co., of 17,300 b.h.p. A contract was placed in 1957 by the East Asiatic Co. for a six-cylinder engine of these new dimensions, of 10,200 b.h.p., to be installed this year in a cargo liner.

Although no such orders have yet been announced, the Harland-built opposed-piston design of B. and W. engine could be built to give 20,000 b.h.p. from eight cylinders with a bore of 880 mm. and a total weight of about 950 tons. With an increased degree of pressure-charging, this could give 24,000 b.h.p. The highest-powered Harland-B. and W. engine entered service early last year in the Royal Mail Lines' 11,778-ton d.w.c. cargo liner *Loch Loyal*. This engine has seven cylinders with a nominal bore of 750 mm. and piston strokes of 1,500 mm. (upper) and 500 mm. (lower), the rated output being 10,300 b.h.p. at 110 r.p.m. Harland and Wolff hold an order at present for a 12,500-b.h.p. turbocharged opposed-piston eight-cylinder engine, to be installed in a 32,000-ton tanker for John I. Jacobs, also for a 42,000-ton tanker with a 15,000-b.h.p. 10-cylinder engine. The owners are Hunting and Sons, Ltd.

THE SWING FROM TURBINE TO DIESEL

One of the most significant trends in 1958 was the switching of contracts from steam turbine-propelled vessels to Diesel propulsion, probably the most outstanding example of this being the decision, announced in our July issue, of London and Overseas Freighters, Ltd., one of Britain's largest independent tanker operating concerns, to adopt Diesel propulsion for two 32,000-ton d.w. turbine tankers ordered from Konink. Mij. De Schelde. Instead of turbines, each vessel will now be propelled by a type 9RD90 single turbocharged De Schelde-Sulzer engine (Figure 1) with the new large-size cylinder (900 mm. bore x 1,500 mm. piston stroke). This has a maximum output of 16,515 b.h.p. (metric) at 119 r.p.m. The service output will be 14,850 b.h.p. at 115 r.p.m. (90 per cent full load) with a corresponding mean effective pressure of 6.55 kg./cm.^{2} (96 p.s.i.) and a mean piston speed of 5.94 m./sec. (1,170 ft./min.).

ENGINES WITH 900 MM. CYLINDER BORE

This RD Sulzer design has only been announced during the past year and succeeds the well-known RSAD76 unit. One of the main modifications in the new engine, whether of the 760 mm. or 900 mm. bore, is that it will now have symmetrical columns, made possible by the elimination of the scavenge pumps. Whereas the framework previously incor-

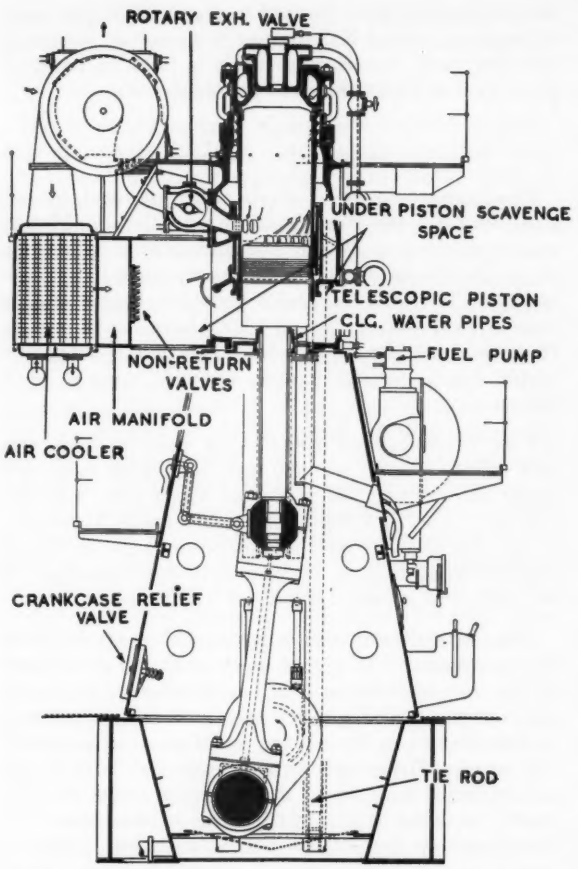

Figure 1. Section through one cylinder of the new design of Sulzer RD90 engine, with a cylinder bore of 900 mm.

porated the scavenging air manifold, this manifold is now a separate unit bolted to the cylinder blocks below the three turboblowers. The scavenging air manifold also incorporates the air coolers. With the 900-mm. bore engines the pistons will be water-cooled, but oil- or water-cooling can be provided, if required, for the RD76 engines. The new design is of welded steel construction for the frames, bedplates, etc., and the 9RD90 unit has a weight of 738 tons.

The first Sulzer RSAD 10-cylinder engine ran sea trials early in 1958, this being installed in the Japanese tanker *Hoei Maru*, of 33,354 tons d.w.c. The engine is designed to develop 13,000 b.h.p. in service at 119 r.p.m., with a b.m.e.p. of 7 kg./cm.^{2} In the past year British-built turbocharged Sulzer engines have run shop trials, the first being a Clark-Sulzer seven-cylinder RSAD76, capable of developing in service 8,750 b.h.p. at 115 r.p.m., and now in service in the tanker *Blyth Adventurer*. A similar engine was recently on testbed trials at the works of Alexander Stephen and Sons, Ltd., before installation in the 15,500-ton tanker for B.P. Tankers, Ltd.

Sulzers' TD48 trunk-piston two-stroke unit has been redesigned during 1958, mainly with the object of equipping it with lateral scavenging air pumps, thus reducing the length. This is a relatively slow-running engine covering a power range of from 1,500 b.h.p. (non-turbocharged) to 4,600 b.h.p. in turbocharged form.

Of particular interest at the present time is the machinery installation selected by Manchester Liners, Ltd., for two 6,000-ton cargo ships, the first of which will shortly be running trials. This vessel has the first turbocharged trunk-piston Sulzer engine to be built in Great Britain.

One of the highest-powered marine Diesel engines will soon be running shop trials at the Glasgow works of Alexander Stephen and Sons, Ltd., this being a Sulzer-type 12-cylinder turbocharged unit rated at 15,600 b.h.p. (14,700 b.h.p. in service). It will be installed in a cargo liner for the City Line (Ellerman Lines).

NEW DOXFORD ENGINE DEVELOPMENT

Two simultaneous but not unconnected lines of development are being vigorously pursued by William Doxford and Sons (Engineers), Ltd., who in the past year have seen their turbocharged opposed-piston engines installed in many classes of ship. At the commencement of 1958 only one turbocharged unit was in sea service, this being the six-cylinder 670-mm. bore unit in the Elder Dempster cargo liner *Egori*, having a rated output of 8,000 b.h.p. at 116 r.p.m. with a b.m.e.p. of 91.2 p.s.i., and a maximum combustion pressure of 760 p.s.i. Turbocharging gives this engine a boost in output of about 20 per cent over that of a corresponding unit.

The year 1958, however, has seen appreciable numbers of turbocharged engines delivered from Doxfords and their licensees for cargo liners, tramps and tankers, the latter including vessels for Shell Tankers, Ltd., who after many years of specifying turbine machinery for powers well within the range of established Diesel engines have again returned to the Diesel type which, until the immediate post-war years, they had supported so well and by their research and experience in this sphere had contributed much towards its present efficiency.

As stated, development is now following two lines: a higher degree of boost over the hitherto very conservative ratings and the construction of a new engine which will be "shorter, lighter and less expensive" than the existing design, although retaining many of the best features. This new unit will provide powers of up to about 16,000 b.h.p. and it is believed that even higher outputs will ultimately be possible. With the new turbocharged units, the reciprocating scavenging air pumps will be dispensed with entirely, as has already been accomplished in the case of the 7,800-b.h.p. Provence-Doxford engine in the 4,900-ton *Fort Niagara*, which entered service in June.

In the recently completed *Thirlby*, a tanker with a Doxford-built turbocharged engine, the scaveng-

ing air pumps have been dispensed with satisfactorily without detriment to the available astern power. Thus, it would appear that the first steps have been accomplished towards supplying all scavenging and charging air by exhaust-gas-driven blowers, as is the case with B. and W., Stork and Sulzer engines.

The trend towards increased percentage boost by turbocharging is also indicated by the Scott-Doxford engine with a bore of 670 mm. Although having the same cylinder dimensions as the first turbocharged Doxford engine, which for 21 months' continuous sea service has given the owners complete satisfaction, it now has a continuous maximum rated output of 9,000 b.h.p. (9,500 b.h.p. has been developed on the test-bed) as against the previous figure of 8,000 b.h.p. Th b.m.e.p. has been increased from the earlier figure of 91.2 p.s.i., to 103 p.s.i.

ADVANCES IN M.A.N. DESIGN

The Maschinenfabrik Augsburg Nürnberg has recently designed a new size of marine Diesel engine with a cylinder bore of 840 mm. and a piston stroke of 1,600 mm. which, with a b.m.e.p. of 7.5 kg./cm.^{2} (106.5 p.s.i.) will give about 20,000 b.h.p. in 12 cylinders. An experimental two-cylinder unit of these cylinder dimensions is now being built by M.A.N. but, with a complete lack of hesitancy which has so characterized Japanese marine Diesel engineering in recent years, the Yokohama Shipyard and Engine Works have already arranged to construct a 12-cylinder engine of this class. It will develop 20,000 b.h.p. in service and will be installed on a 65,000-ton tanker. A 10-cylinder engine of the same type is to be built in Germany for a 47,350-ton tanker.

Over recent years a number of exhaust-gas pressure-charging systems have been adopted for M.A.N. engines, the impulse-series-parallel system being accepted generally as the most efficient for the KZ series, of the higher powers. In this, the undersides of all pistons are used as scavenging air pumps, about two-thirds of them operating on the series system. Automatic disc-type non-return valves are interposed between the first and second stages of the divided scavenging air belt, to open as soon as the capacity of the piston undersides is exceeded by the amount of air delivered by the blowers. The remaining one-third of the under-pistons operate in parallel and draw air from the atmosphere through air coolers directly into the second stage of the scavenge belt. In one or two recent instances, however, all the underpistons operate in series.

The former engine-driven double-acting scavenge pump at the forward end of the engine is now dispensed with, as have other arrangements also with Roots-type blowers, while experience has also shown it to be possible to omit the exhaust-slide valves. Hitherto, the degree of boost given by turbocharging has been limited to about 25-30 per cent, although standard KZ engines have run at boosts of 45.0 per cent, but design work is now in hand to give greater boosts than 30 per cent.

LARGER FIAT ENGINES

The highest-powered single-acting two-stroke engine built by the Fiat Stabilimento Grandi Motori was completed in March last and is now in service in the 31,000-ton tanker *Sicilmotor*, owned by Sicilnaviglio. This is a turbocharged 12-cylinder engine with a maximum output of 18,000 b.h.p. at 145 r.p.m. (8.0 kg./cm.^{2} b.m.e.p.) while the rated output is 14,400 b.h.p. at 132 r.p.m., with a b.m.e.p. of 7 kg./cm.^{2}

Fiat turbocharged engines at present comprise two cylinder sizes; the 680-mm. unit with each cylinder developing 930 b.h.p. at 140 r.p.m. and the 750-mm. type, of 1,200 b.h.p. at 132 r.p.m. However, work is at present well advanced on a two-cylinder experimental engine with a cylinder diameter of 900 mm. and a piston stroke of 1,600 mm.

The two-cylinder unit will be on shop trials about February-March this year. Almost at the same time as the two-cylinder unit is being tested, a nine-cylinder engine will be under construction to the order of Achille Lauro S.p.A., for installation in a 35,000-ton tanker. The engine will be rated at 1,700 b.h.p. per cylinder in service at 118 r.p.m., and will be ready for shop trials by the end of this year, for installation in the ship in February-March, 1960.

The 12-cylinder 14,400-b.h.p. engine in the *Sicilmotor* is the first of welded construction for a merchant ship, is cross-scavenged and has a double-acting scavenging pump for each cylinder working in series with three turboblowers. Constant-pressure turbocharging has been adopted, the percentage boost over the output of a corresponding normally aspirated engine being 40 per cent, the m.e.p. being 7 kg./cm.^{2}

NOHAB-POLAR TURBOCHARGED ENGINE IN BRITISH SHIPS

Two British ships have entered service in the past year propelled by Swedish-designed and built Polar two-stroke engines, each of these vessels, owned by F. T. Everard and Sons Ltd., London, having a turbocharged-type MN16S unit built by Nydqvist and Holm A/B, Trollhättan, and giving a maximum continuous service output of 1,235 b.h.p. at 250 r.p.m. The output represents a 30 per cent boost over that of a corresponding non-turbocharged unit, the mean indicated pressure at 100 per cent load (1,410 i.h.p.) being 116.5 p.s.i. Turbocharging is on the impulse system, dispensing with the Nohab-Straatveit ring-piston double-acting pump of the non-turbocharged units.

With a length of 20 ft., and a total weight, includ-

ing pumps, of 42 tons, this engine has a guaranteed fuel consumption of 162 gr. (0.356 lb.) per b.h.p.-hr.

SPECIALIZATION ON ONE ENGINE

The highest-powered ship with Ruston engines entered service in 1958, this vessel, the 15½-knot twin-screw short-sea passenger-cargo liner *Bolton Abbey*, and her sister-ship now nearing completion, each being propelled by two 8VOXM engines driving through a reverse-reduction gearbox and giving a continuous rated output of 1,806 b.h.p. at the tail-shaft (a total of 3,612 b.h.p.) at 109 r.p.m. A point of particular interest is that the owners, Associated Humber Lines, Ltd., have specified similar VOXM engines, but with seven-cylinders, for six cargo ships on order, each of which will develop 1,580 s.h.p. in service to give a speed of 12½ knots. In each case, the engine will be unidirectional and drive through a Renk reverse-reduction gearbox.

These owners have thus standardized on one class of engine for their fleet and when the series of six ships is completed, will have 10 vessels propelled by VOXM engines, two of them with twin screws. Ruston auxiliary Diesel engines have also been specified for these vessels to enable spares to be interchangeable between ships, and to reduce the amount of capital locked up in spare components. Such an arrangement also enables engine-room staff to be conveniently and safely switched to other vessels and be fully acquainted with the machinery.

The VOXM engine is a pressure-charged four-stroke unit with a bore of 17 in. (432 mm.), a piston stroke of 18 in. (457 mm.) and a b.m.e.p. at maximum output of 119.2 p.s.i.

BRITISH SHIP WITH GERMAN GEARED-DIESEL INSTALLATION

Another British ship with non-British machinery and of special interest, with twin Deutz Diesel engines coupled to a single shaft, is the Buries Markes ore carrier *La Colina*, built in France and of 10,000 metric tons d.w.c. Designed for a service speed of 12 knots, this vessel has a 4,000-b.h.p. propelling installation occupying an overall floor area (including gearing and clutches) of 49 ft. by 16.4 ft. with a height of only 9.8 ft. Thus there is a trend towards engines which hitherto have been considered by many as suitable for only the smaller classes of vessel to be selected for relatively large sea-going ships, the Deutz engines in this case being eight-cylinder, turbocharged four-stroke units with a cylinder bore and stroke of 420 mm. and 660 mm. respectively, and an output of 2,000 b.h.p. at 275 r.p.m. Each weighs about 52 tons.

Another multi-engined installation is the recently completed *Archsum*, of 4,600 tons d.w., for the Nordforesische Reederei, Rendsburg, and equipped with Diesel-electric plant. The propulsion motor is supplied with current from five D.C. generators, each driven at 1,300 r.p.m. by a Mercedes-Benz 885-b.h.p. engine. There are also three engines of similar make (two of 285 b.h.p. and one of 105 b.h.p.) for supplying current for lighting and other duties.

Of particular interest is the bulk cargo carrier *Bahama King*, formerly the 11,179 gross tons motor tanker *Geneve*, now returning from her first voyage since being re-engined with Napier Deltic high-speed opposed-piston Diesel engines. She is, in fact, the first merchant ship to be propelled by such machinery, although the Shell Co., for example, has adopted these units for numbers of fast passenger-carrying launches, and they have also been used extensively on Naval vessels.

The original two Krupp Diesel engines in the *Bahama King*, totalling 2,600 b.h.p., have now been replaced by four 18-cylinder Deltics, connected in pairs to each propeller shaft. The continuous rating of each is 1,725 b.h.p. at 1,500 r.p.m. and in service the total output will be about 5,300 s.h.p. The weight/power ratio is 5.27 lb. per b.h.p.

Pielstick high-speed engines are being specified to an increasing extent for various classes of ship, but probably the most interesting installation in 1958 was that adopted for the French National Railways car ferry *Compiègne*, with 9,000-b.h.p. machinery, giving a speed of 20 knots.

FUTURE PROPELLING MACHINERY

Having briefly summarized a few of the more outstanding marine Diesel engineering requirements of the past year, what lies ahead? This year will see the shop trials of at least three large-bore marine Diesel engines, generally of the builders' standard design; Doxfords will run trials of their new class of engine; percentage boosts by turbocharging will be stepped up and two free-piston-engined merchant ships will be in service. So will the *Auris*, at last re-emerging in her new role of gas-turbine-propelled tanker.

Of the five free-piston gas-turbine installations ordered for ships in Great Britain, one was cancelled in the past year, this being the 4,000-b.h.p. installation ordered from Alexander Stephen and Sons, for a ship of the British-India Steam Navigation Co. It was decided by the owners not to proceed with this conversion, from existing steam plant, in view of the condition of the freight market.

Within the next few weeks the first merchant ships to be engined with free-piston plant in Great Britain will be running sea trials, one of these being the new ore carrier of about 9,400 tons d.w.c. ordered from Lithgows, Ltd., by Scottish Ore Carriers, Ltd. (J. and J. Denholm, Ltd., managers) and this is being propelled by two GS34-type gasifiers, built by the Free-Piston Engine Company, and which supply gases to the 2,500-s.h.p. gas turbine built by Rankin and Blackmore to the designs of Power Jets (Research and Development), Ltd. This turbine develops its output of 2,500 s.h.p. at 5,250 r.p.m. when running in the ahead direction, and drives through gearing of the articulated locked-

train double-reduction type built by the Fairfield S.B. and Eng. Co.

Also being fitted out at the present time is the former steamship *Goodwood*, owned by France, Fenwick and Co., Ltd., of 3,530 tons d.w.c., which was built in 1949 and propelled by triple-expansion steam machinery. This machinery is now being replaced by a 2,000-s.h.p. free-piston installation, the principal engineering contractors being Smith's Dock Co., Ltd.

The first ship to have a combination of free-piston machinery and a variable-pitch propeller will be the 12,500-ton d.w. cargo ship ordered by the Bolton Steamship Co., Ltd., from Smith's Dock Co. This will have four Smith-built gasifiers and a 4,000-s.h.p. B.T.H. gas turbine driving a KaMeWa V.P. propeller ordered from J. Stone and Co.

The year 1958 has been disappointing so far as the construction and development of the 5,500-s.h.p. gas turbine for installation in Shell Tankers' *Auris* is concerned. This ship, which is equipped with an open-cycle turbine, has not yet been placed in service.

Thus, in the gas turbine field, there will be two free-piston-engined cargo ships and one gas-turbine-propelled tanker entering service within a few weeks. Marine orders have not progressed for such plant over the past year or so, nor is it likely that they will do so until other owners can benefit from the experience of those who have preferred to take the initiative. The early praises of the gas turbine are today faint, and the promises of a soon-forthcoming, simple successor to the marine Diesel engine ring empty.

An order has been placed with the Mitsubishi Nippon Heavy Industries, Ltd., for a 65,000 DWT single screw tanker to be propelled by a twelve cylinder Diesel engine having a 20,000 bhp continuous rating. The engine will be built under license from the M.A.N. works in Germany and will be the largest ever constructed. It will make its rated power at 112 to 115 rpm. Its bore will be nearly 33 inches and the stroke about 63 inches. A two-stroke, supercharged type, it will have a piston speed of about 1200 fpm.

—from THE MARINE ENGINEER AND NAVAL ARCHITECT, January, 1959

A 2500 shp free piston gas turbine installation has been completed in the British ship MORAR. The first of its kind in the British merchant marine, the installation consists of three GS-34 gasifiers, each designed to deliver about 1000 hp and supply gas to a Rankin and Blackmore gas turbine. The turbine is rated at 2500 shp and 5250 rpm. Designed to use bunker fuel, the plant showed a consumption of about 0.49 lb/shp-hr on trials. It is expected that this will be reduced by improvements to be made, a value of 0.42 lb/shp-hr being the present target.

—from THE MOTOR SHIP, February, 1959

IRVING GRANET

SOME NOTES ON BOILING HEAT TRANSFER

THE AUTHOR

was born July 2, 1924. Attended Brooklyn Technical High School and received the degree of Bachelor of Mechanical Engineering from the Cooper Union in 1944. Employed in 1944-45 on Navy Torpedo project as project engineer on stress and vibration studies. Subsequently served two years with Air Corps overseas as Flight Engineer. In June 1948 received the degree of Master of Mechanical Engineering from the Brooklyn Polytechnic Institute and in 1949 received Professional Engineering License in New York. Since 1947, employed by Foster Wheeler Corporation as Project Engineer, Research Department; then Director of Staff Engineering, Nuclear Energy Department. In 1955-56 attended the Oak Ridge School of Reactor Technology at Oak Ridge, Tennessee. Has taught Heat Transfer and Thermodynamics at Brooklyn Polytechnic and is presently completing the requirements for the degree of Doctor of Mechanical Engineering. Has published over twenty articles on gas properties, heat transfer thermodynamics and applied mechanics. Recently was Consultant, Nuclear Energy Department, Foster Wheeler Corporation, New York. Is now on the Scientific Research Staff of Republic Aviation Corporation.

IN A PREVIOUS paper (1), the writer indicated that the literature on boiling is relatively extensive, but leaves much to be desired before it is useful to the designer. A recent paper by Levy (2) correlates these data and "Good agreement between test results and the derived equation was obtained for pool boiling and nucleate boiling heat transfer of sub-cooled and vapor containing liquids." The approach utilizes a simplified model of the boiling mechanism close to the heated surface to obtain a generalized heat transfer correlation. By comparing equations (8), (11) and (12) in Levy's paper for saturated pool boiling, nucleate boiling with net steam generation and sub-cooled nucleate boiling, respectively, it will be found that all of these correlative equations are similar except for a factor in each case. This represents a simplification since one need only evaluate one equation and apply the necessary corrections.

Equation (1) below is Levy's equation (8) for pool boiling at saturation temperature:

$$\frac{Q}{A} = \frac{k_L C_L \rho_L^2}{\sigma T_s (\rho_L - \rho_V)} \frac{1}{B_L} (\triangle T)^3 \ldots\ldots\ldots\ldots (1)$$

The coefficient B_L is an empirical function of fluid properties. In particular, it is found that B_L is a function of $\rho_V h_{fg}$. The value of $\frac{1}{B_L}$ is given in Figure 1 as taken from Levy's paper. This curve combined with equation (1) above . . . "is of a generalized nature applicable to all considered fluids from low to very high pressures, and independently of heating surface-fluid combination." The conclusion presented above is quite startling since it is known that boiling heat transfer increases with increased surface roughness, properties of fluids are difficult to obtain at high pressure and that the

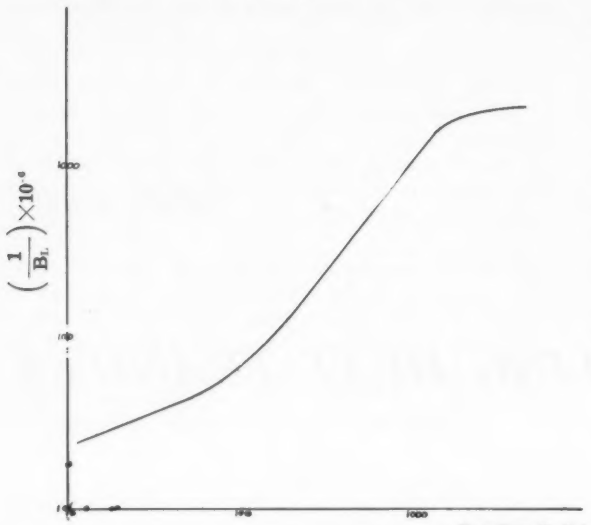

Figure 1. Empirical Coefficient for Nucleate Boiling—(Equation 1).

normal spread of experimental data in the boiling field is quite high.

Considering equation (1), it is possible to express the boiling heat transfer coefficient as follows:

$$\frac{Q}{A(\triangle T)}=h=\frac{k_L C_L \rho_L^2}{\sigma T_s(\rho_L-\rho_V)}\left(\frac{1}{B_L}\right)(\triangle T)^2 \ldots\ldots (2)$$

Since every term on the right hand side of equation (2) is a unique function of saturation properties, one can rewrite equation (2);

$$h=F(\triangle T)^2 \ldots\ldots (3)$$

where F is the coefficient of correlation and includes B_L. It will be noted that F has physical dimensions. In order to evaluate F for water it is necessary to evaluate the surface tension, specific heat, density and thermal conductivity as functions of pressure and temperature. The data for specific heat, density, temperature and latent heat of evaporation can all be obtained from the Steam Tables (3). For surface tension the writer has used the data from reference (4) extended over the pressure range linearly to zero at the critical point as suggested by Rohsenow reference (5). This is in reasonable agreement with the data given by Gambill (6). The thermal conductivity is from reference (8), extended linearly as required. These data are shown graphically for water in Figure 2. Table 1 is the work sheet for the evaluation of F. These calculations were carried out for water up to 2,000 psia and estimated to 2,500 psia. This would appear to be the limit at present with the data at hand.

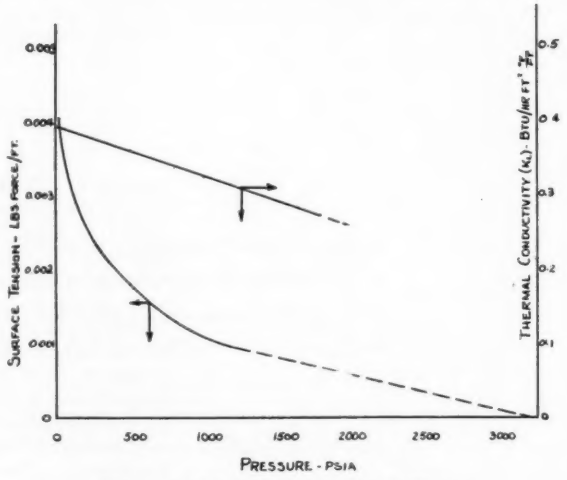

Figure 2. Selected Properties of Water.

The most useful presentation of F would be as a function of temperature. This would require a knowledge of the pressure-temperature relationship for the saturated liquid. Rather than this, it is relatively easy to plot F as a function of pressure and convert to temperature by use of the Steam Tables. Figures 3 and 4 show this plot. It will be noted from these curves that for a given $\triangle T$, h increases with pressure. From Figure 4, it is further noted that F plots linearly on log-log paper over the range from 100 psia to 2000 psia. Therefore, over this range, it

TABLE I—*Evaluation of "F" for Water*

Pressure Psia	t_s °F	T_s °R	ρ_L Lbs/ft³	ρ_V Lbs/ft³	$\rho_L-\rho_V$ Lbs/ft³	C_L Btu/lb °F	K_L Btu/hr ft °F	σ lbs/ft	h_{fg} Btu/lb	$\rho_V \cdot h_{fg}$ Btu/ft³	$\left(\frac{1}{B_L}\right)\times 10^{-6}$	F
14.7	212	672	59.8	.037	59.76	1.005	.394	.00404	970.3	35.9	38	0.257
50	281.01	741.0	58.0	.117	57.88	1.025	.390	.00355	924.0	108.5	70	0.48
100	327.81	787.8	56.4	.226	56.17	1.040	.388	.0031	888.8	200.0	130	0.94
250	400.95	861	53.7	.542	53.16	1.085	.380	.0024	825.1	447.0	370	3.12
500	467.01	927	50.70	1.10	49.60	1.150	.360	.0018	755.0	829.0	780	7.78
800	518.23	978.2	47.90	1.76	46.14	1.225	.338	.0013	688.9	1212.0	1300	16.30
1000	544.61	1004.6	46.30	2.24	44.06	1.282	.324	.00108	649.4	1451.0	1600	23.20
1250	572.42	1032.4	44.50	2.90	41.60	1.375	.306	.00092	602.4	1748.0	1800	29.5
1500	596.23	1056.2	42.55	3.62	38.93	1.500	.292	.00080	556.3	2015.0	1900	35.8
2000	635.82	1095.8	38.90	5.33	33.57	1.790	.258	.00058	463.4	2470.0	2010	51.0

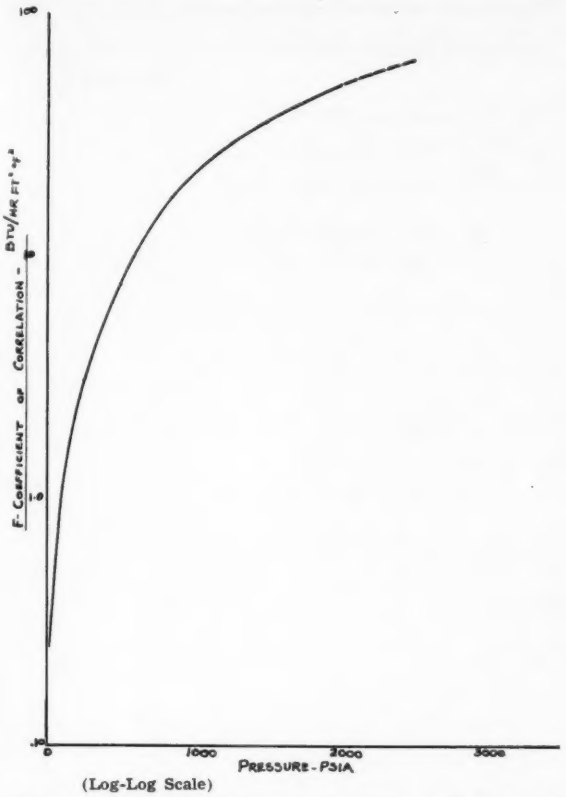

Figure 3. Coefficient of Correlation "F" as a Function of Pressure.

is possible to express F mathematically as:

$$F=\frac{P^{4/3}}{495} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (4)$$

The utility of the curves and equations developed above can best be shown by an example. In this example, the first assumption will be that the steam produced does not effect the boiling coefficient. This will subsequently be reviewed and the net vapor generation will be considered. From reference (1), we have a typical problem of this type:

"Size a single pass, straight tube steam generator with the following information given:

Primary Water:

Flow rate (W) 2.5×10^6 lbs/hr.
Temperature entering steam generator—520° F
Temperature leaving steam generator—480° F
Operating Pressure—2000 psia
Design pressure drop (not including nozzles) 12 psi

Secondary

Steam generated at 400 psia (444.6° F)

Tubing

Type 347 stainless steel, seamless, design pressure 2500 psi, design temperature 600° F—Outside diameter ½"; spacing ¾" triangular centers."

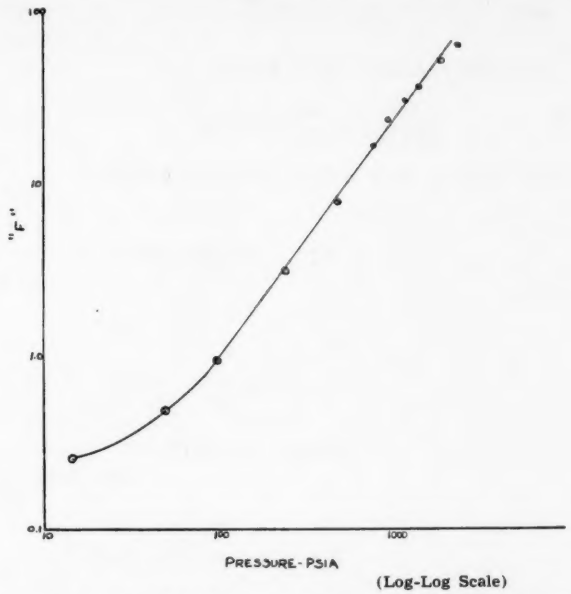

Figure 4. Coefficient of Correlation "F" as a Function of Pressure.

"The minimum thickness of tubing required can be determined from the information given. It will be assumed that the steam side design pressure will not govern. Calculating the minimum thickness based upon internal pressure yields a value of 0.0393 inches. The nearest standard thickness would be 0.042 inches, and because of tolerance, this would be a minimum. It is possible to purchase tubing to other than standard gauges without incurring special charges (in some instances), but for the present, it will be assumed that the actual tube wall is 0.042". Therefore, the inside diameter is 0.416 inches."

One remark should be made at this point; in all of the foregoing, no mention has been made of the orientation of the surface, i.e., horizontal or vertical unit. At present there would seem to be a lack of information on this point, but it appears (7) that a single vertical tube may not give boiling film coefficients as high as single horizontal tubes. This difference can not be given a mathematical representation and will not be considered here.

The surface required for the unit can be obtained by integrating the heat balance equation. The validity of this approach can be questioned since the process of integration along the tube involves certain assumptions which may not be physically realized in a heat exchange. However, this is the only procedure available with the data at hand. Therefore:

$$A=WC_L\int\frac{d(\triangle T_f)}{U(\Delta T_f)} \quad \ldots\ldots\ldots\ldots\ldots\ldots (5)$$

with $$\Delta T_f = \left(\sum_1^n R\right)\frac{Q}{A} \quad \ldots\ldots\ldots\ldots\ldots\ldots (6)$$

The tube resistance is as before:

$$\frac{0.25}{12\times10^{-6}}\ln\frac{.25}{.208} = .000369$$

The outside scale will be taken = .000333

The inside scale will be neglected

The inside (water) resistance is $= \dfrac{1}{1.48\times10^6\left(\frac{1}{N}\right)^{-8}}$

For this pressure, F = 6

Therefore $h = 6(\Delta T)^2$

and

$$\Delta T_f = \left[.000369 + .000333 + \frac{1}{1.48\times10^6\left(\frac{1}{N}\right)^{-8}}\right]\frac{Q}{A} + \frac{1}{6^{1/3}}\left(\frac{Q}{A}\right)^{1/3} \quad \ldots (7)$$

or

$$\left.\begin{aligned}\Delta T_f &= (D)\frac{Q}{A} + \frac{1}{6^{1/3}}\left(\frac{Q}{A}\right)^{1/3}\\ \text{where } D &= .000702 + \frac{1}{1.48\times10^6\times\left(\frac{1}{N}\right)^{-8}}\end{aligned}\right\} \ldots\ldots\ldots (8)$$

Therefore $$d(\Delta T_f) = Dd\left(\frac{Q}{A}\right) + \frac{1}{6^{1/3}}\left(\frac{1}{3}\right)\left(\frac{Q}{A}\right)^{-2/3} d\left(\frac{Q}{A}\right) \ldots (9)$$

Placing (9) into (5) yields the following:

$$A = WC_L\int \frac{Dd\frac{Q}{A} + \frac{1}{6^{1/3}}\left(\frac{1}{3}\right)\left(\frac{Q}{A}\right)^{-2/3} d\left(\frac{Q}{A}\right)}{\frac{Q}{A}} \quad \ldots\ldots\ldots (10)$$

Integrating (10) between limits:

$$A = WC_L(D)\ln\frac{(Q/A)_1}{\left(\frac{Q}{A}\right)_2} + WC_L(.275)\left[\left(\frac{Q}{A}\right)_2^{-2/3} - \left(\frac{Q}{A}\right)_1^{-2/3}\right]. \quad (11)$$

Based upon pressure drop considerations, it is possible to write an expression relating the number and length of tubes. Reference (1), equation (1a) gives this expression for this example as:

$$\left.\begin{aligned}N &= 0.315\times10^3\sqrt{.404L + 1.904}\\ \text{or}\quad L &= \frac{N^2}{40{,}000} - 4.73\end{aligned}\right\} \ldots\ldots\ldots\ldots (12)$$

Noting that $A = \frac{\pi}{24}NL$, equation (11) becomes:

$$L = \frac{24}{\pi N}\left[WC_L(D)\ln\frac{(Q/A)_1}{(Q/A)_2} + (.275)(WC_L)\left[\left(\frac{Q}{A}\right)_2^{-2/3} - \left(\frac{Q}{A}\right)_1^{-2/3}\right]\right. \quad \ldots\ldots (13)$$

By successive application of equations (7), (12) and (13) it is possible to arrive at the desired solution. The procedure is as follows:

1. Assume a value of N.
2. Place this value in equation (7) to give ΔT_f as a function of (Q/A).
3. Plot the above function.
4. For the values of ΔT_f at inlet and outlet $(Q/A)_1$ and $(Q/A)_2$ can be found.
5. Evaluate L from equation 13.
6. Evaluate L from equation 12.
7. Repeat to obtain enough points for 2 curves. Plot L versus N.
8. The intersection of these curves gives the desired solution.

To illustrate this procedure the following points on each curve will be calculated:

1. N=1000 Tubes
2. $\Delta T_f = [.000871]\frac{Q}{A} + 0.55\left(\frac{Q}{A}\right)^{1/3}$

3. Figure 5 is a plot of ΔT_f vs. $\frac{Q}{A}$ for this case

4. @ $\Delta T_f = 75.4, \left(\frac{Q}{A}\right)_1 = 62{,}000$ and @ $\Delta T_f = 35.4, \left(\frac{Q}{A}\right)_2 = 22{,}500$

5. From Equation (13), L=22.4 feet
6. From Equation (12), L=20.17 feet

Repeating the above:

1. N=900 Tubes

2. $\Delta T_f = (.000857)\frac{Q}{A} + 0.55\left(\frac{Q}{A}\right)^{1/3}$

3. On Figure 5, the above has been plotted.

4. @ $\Delta T_f = 75.4, \left(\frac{Q}{A}\right)_1 = 63{,}700$ and @$\Delta T_f = 35.4, \left(\frac{Q}{A}\right)_2 = 23{,}000$

5. From Equation (13) L=24.7 feet
6. From Equation (12) L=15.52 feet

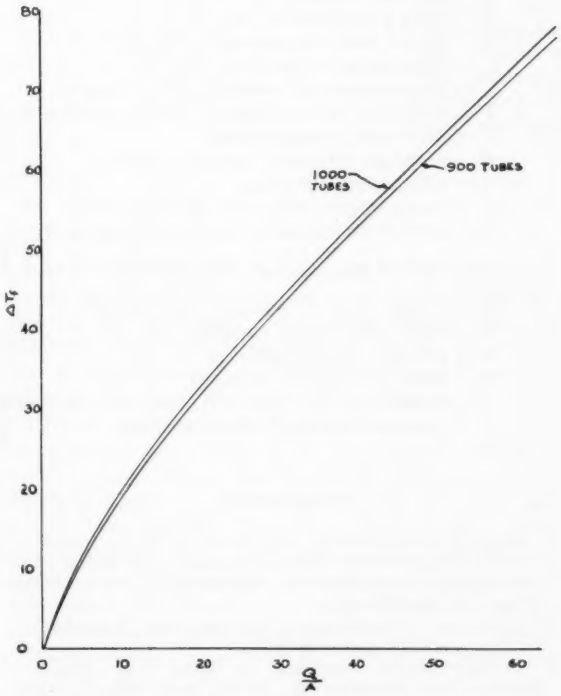

Figure 5. ΔT_f vs. $\frac{Q}{A}$ for Assumed Tube Counts.

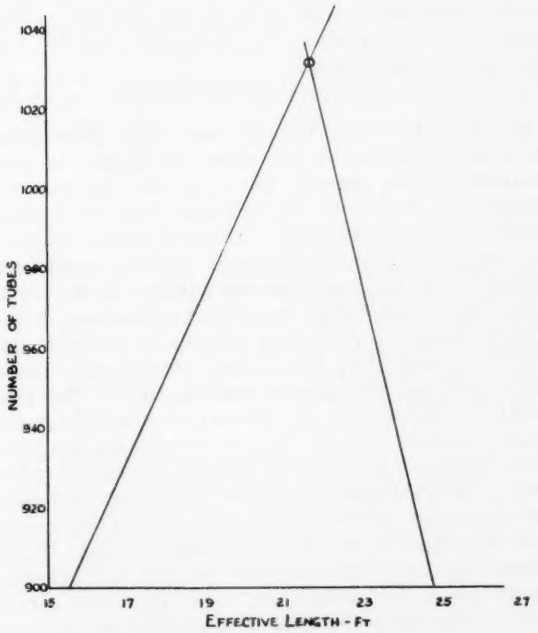

Figure 6. Solution of Illustrative Problem.

Plotting the above yields figure 6, with the solution being 1031 tubes of 21.65 feet effective length. This corresponds to an effective surface of 2930 sq. ft. or a "U" overall of 757. The corresponding values from reference (1) using 2000 for boiling plus scale are 2350 and 965 respectively.

It is also apparent at this point that a small increase in the steam side operating pressure could increase the boiling coefficient considerably. This phenomenon of an increased peak heat flux and increased boiling coefficient with increased pressure has been reported elsewhere in the literature (reference 8) and it is noted that at 2465 psia R has been found to be approximately 100 times the value at atmospheric pressure.

For completeness, it is necessary to briefly discuss the cases of sub-cooled nucleate boiling and nucleate boiling with net steam generation. For sub-cooled nucleate boiling Levy gives an equation of the same form as equation (1), multiplied by a correction factor for sub-cooling. For water this

correction factor can be expressed as:

$$\left[1+\frac{(\text{degrees subcooling})}{h_{fg}}\right] \quad \ldots\ldots\ldots\ldots (14)$$

Under most conditions this correction factor is relatively small (say 10-20 per cent) and it must be remembered that it is applied to a non-controlling resistance. In most cases, feed is usually introduced into the unit in such a manner that most of the sub-cooling is removed by direct contact with steam. This is usually done in the steam drum or separation volume in the unit. Under these conditions the amount of sub-cooling is very small in the downcomer section and it is the writer's considered opinion that it should be neglected in design. At most, this represents a very small additional margin of safety in design and introduces a great deal of difficulty for the designer which the data do not justify.

Nucleate boiling with net heat generation is probably the most common case to be found in industry. Again, Levy derives an equation for this case of the same form as equation (1) multiplied by the following correction factor:

$$[1-\text{Fraction Steam by Weight}] \quad \ldots\ldots\ldots (15)$$

The particular correction for net steam generation as given by equation (15) does not appear to correspond to the general literature on the subject. McAdams (reference 8) indicates that the boiling coefficient increases with increasing steam fraction by weight and then decreases due to insufficient liquid to wet the heat transfer surface. In the limit, the value reaches that for forced convection of the vapor. Equation (15) goes to zero at 100 per cent vapor. Apparently, the author considered boiling only, divorced from any convective effects. This procedure of separating the convective effects from the boiling effects may be fine for correlative purposes but it does not give the designer the necessary information he must have. Therefore, it becomes necessary to once again make a judicious conservative choice for this case. It would appear that the data in the literature shows this marked decrease in h for fractions steam by weight of 60-80 per cent. These weight fractions correspond to circulation ratios of 1.6 and 1.25 respectively and represent values that would probably not be used except in certain very critical cases. The more usual value for the circulation ratio would be approximately 5 or higher, representing 20 per cent or less steam by weight at the exit of the boiling section. Since the data available in the literature are qualitative and since it is apparent that Levy's correlation does not yield the required information, the writer suggests neglecting this effect at present. If the design is relatively "tight," the effect of sub-cooling and nucleate boiling with net steam generation can be roughly approximated by multiplying "F" by a factor depending on the design parameters. The effects of sub-cooling and net steam generation tend to compensate each other but it is not possible to give any definitive rules at the present state of the art.

NOMENCLATURE

A = heating surface area, ft^2
B_L = coefficient, nondimensional
C_L = specific heat of the fluid, Btu/lb deg F
h = bubble heat-transfer coefficient, Btu/hr ft^2 deg F
h_{fg} = latent heat of evaporation, Btu/lb
k_L = thermal conductivity of saturated liquid, Btu/hr ft deg F
Q = heat-transfer rate from heating surface, Btu/hr
$\triangle T$ = temperature difference equal to $T_w - T_s$, deg R
μ_L = viscosity of liquid, lb/hr.ft
ρ_L = density of liquid, lb/ft^3
ρ_V = density of vapor, lb/ft^3
σ = surface tension, Btu/ft^2
T = fluid temperature, deg R
T_b = mixed bulk temperature, deg R
T_s = saturation temperature, deg R
T_w = temperature of heating surface, deg R
F = coefficient of correlation, Btu/hr ft^2 deg F^3
t_s = saturation temperature, °F
P = absolute saturation pressure psia
$d(\ )$ = differential operation
$\triangle T_f$ = temperature difference between fluids, °F
U = overall heat transfer coefficient Btu/hr ft^2 °F
$\sum_1^n R$ = sum of the thermal resistance from 1 to n
W = weight flow of heating fluid, lbs/hr
D = value defined by equation (8)
L = effective tube length, ft.
N = number of tubes to carry
Subscripts (1) and (2) refer to inlet and outlet heating fluid conditions.

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LIEUTENANT COMMANDER BURTON I. EDELSON, USN

THE 200-SERIES STAINLESS STEELS

THE AUTHOR

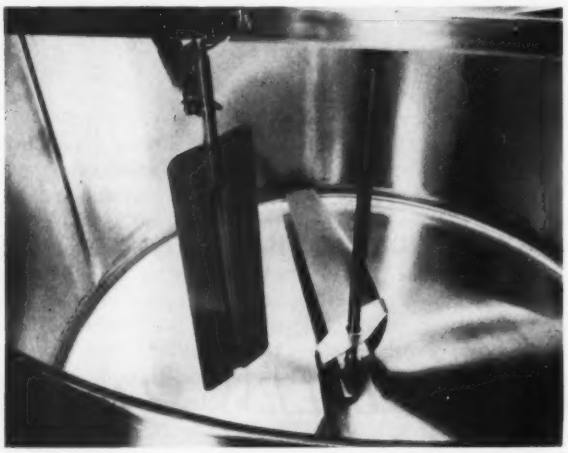
graduated from the Naval Academy in 1947. He attended the Naval Postgraduate School in Monterey and holds a M.S. degree in Metallurgy from Yale University. He served aboard ships in both the Atlantic and Pacific Fleet and is currently assigned as Planning Officer, Resident Supervisor of Shipbuilding, Cleveland, Ohio. His last article, "Science and Scientists in the Navy," was published in the November 1957, JOURNAL.

OVER TWO YEARS ago there occurred an exchange of correspondence between two very notable gentlemen about a seemingly routine subject. The notable gentlemen were the Honorable Frank D. Newbury, then Assistant Secretary of Defense, and Mr. Benjamin F. Fairless, President of the American Iron and Steel Institute (AISI), and the subject was the development of a series of low nickel stainless steels. It seems that the so-called "200 Series" of Stainless steels had just been officially standardized by the AISI and Mr. Newbury had written a quite complimentary note to commend the AISI for its efforts. Whereupon, Mr. Fairless, accepting with agile grace the Secretary's congratulations for the AISI's work in developing these new and useful stainless steels, tossed the ball squarely back into the hands of Mr. Newbury and his Defense Department for the next play. It was, according to Mr. Fairless, not sufficient for research organizations to develop a new material, someone had to take the initiative in specifying applications. In this case, as often times in the past, the mighty defense users could most readily start the ball rolling with this totally new series of important fabricating materials. *The next major contribution to the use of lower stainless steels,* Mr. Fairless wrote, *must come from your own department. Until your people change their material specifications to permit the use of these steels in appropriate applications, your suppliers and others who copy your lead in their own specifications, will, necessarily, have to use the higher nickel grades. This action, in fact, could be taken with respect to other government specifications under which high nickel steels are required. There are many instances where such steels are used when the end-use performance does not require them.*

The complete texts of these letters appear in the Appendix. The sentences above are quoted to underscore the real theme of the present article. Although an attempt will be made to identify and describe briefly this particular series of stainless steels, it is intended to emphasize that the 200 series is only one example of new materials currently available. Many such materials require only recognition in specification writing to obtain extensive use. Alert and enlightened engineers can now put such products to use. These steels represent a particularly cogent example, for heavy industry product development depends almost completely on acceptance by the really large consumers. The Defense Department is certainly pre-eminent in this category. The entire marine industry, including commercial users, is another very large consumer.

STAINLESS STEELS

Stainless steels, which the Navy and marine industry usually term corrosion-resisting steels (or CRES), may be of many different types.



The production of chrome-nickel-manganese stainless steels, which are harder and have higher yield strength than their 18-8 straight Cr-Ni counterparts, increased sharply from a modest 2000 tons in 1955 to 25,000 tons in 1957. This dairy processing tank, fabricated from Type 202 by Cherry-Burrel Corp., represents a typical "200" series application.

In order to understand the general applications of the various types of stainless steels and to differentiate between them, it is sufficient to know a few basic facts about chemical compositions and resulting physical properties. The most important difference between "stainless" and common steels is in their content of the alloying agent, chromium. This all-important addition imparts corrosion resistance to steel. Steels containing more than 12 per cent Cr are arbitrarily known as "Stainless steels." Actually, this is a real misnomer, as none of these steels are really stainless—they all succumb to certain alien corrosive environments—and, in point of fact, many are not really "steels" at all but actually iron alloys.

Nevertheless, accepting certain existing limitations, metallurgists admit the term "stainless steel" and proceed to divide them into three basic types:

a. Ferritic
b. Martensitic
c. Austenitic

These esoteric names relate to the room temperature crystal structures of the various steels. Ferrite is the ordinary room temperature structure of pure iron. Ferritic stainless steels are principally iron-chromium alloys. They contain only a minimum of carbon. These are the cheapest of the stainless grades. The addition of carbon creates the quench-hardenable martensitic stainless steels. These make excellent pump and valve parts and are the finest surgical and cutlery steels.

Austenite is a crystal structure ordinarily stable at high temperatures only. With the addition of about 7 per cent nickel or more, the austenite range is lowered to room temperature and austenitic stainless steels result. Austenitic stainless steels are more expensive than the others, due mostly to the costly nickel addition, but they have certain marked advantages in that they are:

1. More easily cold worked
2. Much more easily welded
3. More stable at both high and low temperatures
4. Stronger and more ductile at both high and low temperatures
5. Non-magnetic

Austenitic steels are numbered by the American Iron and Steel Institute (AISI) in the 300 series; ferritic and martensitic steels in the 400 series.

The most popular, and by far the best known of the austenitic stainless grades, is "18-8" (18 per cent chromium, 8 per cent nickel) standardized as AISI type 302, a fully austenitic grade. Some other austenitic stainless grades have particular properties suitable for special purposes. Examples are 301, a minimum 17-7 grade; 304 with higher amounts of both Cr and Ni for extra corrosive environments; 347 which is an 18-8 grade stabilized against embrittling carbide precipitation; 310—a 25-20 grade for super corrosion resistance at high temperature, and many others.

INTRODUCING THE 200 SERIES STAINLESS STEELS

A new type of austenitic stainless steel has come into use within the past three years, containing manganese as a substitute for a portion of the nickel. Manganese is an austenitizing agent, like nickel. These high-Mn, low-Ni, fully austenitic stainless steels, are known as the 200-series stainless steels. There are two principal alloys now available known as AISI types 201 and 202, which may be substituted for the common AISI 301 and 302 types, and occasionally, even for types 304 and 347. Special types, such as 204 and 204L, not yet formally classified by AISI, designed for direct substitution for 304 and 304L, are not in mill production.

The standardization of these 200 series stainless steels is the culmination of a development process covering 20 years or more and actual use for over a decade. Although there was some considerable laboratory interest in manganese-bearing austenitic stainless steels in the 1930's, the real impetus for their use came with the critical shortage of nickel during World War II. Whereas nickel was then vitally needed for alloy steels—especially armor plate—and other highest priority uses, manganese, on the other hand, was in plentiful supply. Substantial quantities of a 17Cr-4Ni-6Mn steel were produced to replace type 301 in the transportation industry for use in trailer trucks and railway cars. Many of these still remain in service. During the Korean War, nickel again became an extremely critical scarce item, so much so that its commercial use was drastically curtailed by the M-80 regulation. The paucity and the close governmental control of nickel lasted until 1957—or about the time of the recent recession. During this time, the Cr-

Bradley Washfountain Company, Milwaukee, Wisconsin, has adopted the new low-nickel, Type 202 stainless steel for use in place of Type 302 for large washfountain bowls. Shown in the accompanying photograph are typical units of the Bradley line which includes 54" and 36" circular or semicircular bowls.

In tests made on Type 202 no production changes were required nor were any significant differences between Type 202 and Type 302 observed.

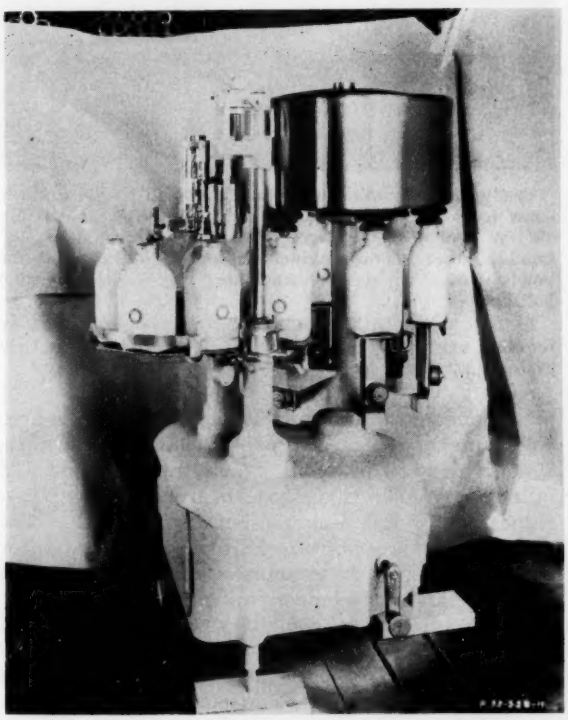
The company is now going into production with Type 202 for the bowls and the center columns which support the spray head.

Mn-Ni steels really caught on and the two grades, 201 and 202, were officially standardized. Great progress was made during this period in understanding their service advantages and limitations, the peculiar problems in their production, and fabrication, and weldability, and strong inroads were made in consumer acceptance. The actual employment of 201 and 202 steels extended to many diverse industries, in most cases replacing 301 and 302 steels in mildly corrosive environments. Perhaps the best example is the transportation industry where the experience gained over the years has promoted type 201 as an acceptable and permanent replacement for type 301 in trailer truck bodies and railway cars. Type 201 wheel covers have been in use on passenger cars. One 1957 luxury automobile has a type 202 roof.

The food processing industry, traditionally a heavy user of austenitic stainless steels—and one quick to suffer from war-caused shortages, was quick to adopt the 200 series. One large brewery has been testing the use of Cr-Mn-Ni stainless for use in beer barrels. A group of experimental barrels kept under close observation while in continuous service between the brewery and a distributor during a two-year period, revealed that these steels were equivalent from the standpoint of corrosion resistance, appearance and denting to the 302 and 304 barrels ordinarily used.

Type 202 has been used on dairy tanks, cream separator disks, milk cans, milking machines and bulk milk coolers for parts not actually in contact with milk. A bulk milk cooler fabricated entirely from type 202 was exhibited at the 1956 Dairy Industries Exhibit. A large orange and grapefruit juicer made of type 201 has been in use for two years. Malted milk cups, syrup containers, decorative panels and soft drink tanks for vending machines are additional examples of employment by food industries.

These steels are well adopted to architectural use. About 200 tons of type 202 are being used for all opaque exterior surfaces on the new Inland Steel building in Chicago. Union Carbide's plans for a new 42 story office building in Manhattan include 202 steel. Allegheny Ludlum—in the forefront among individual commercial developers of Cr-Ni-Mn steels—plans to sheath 3 of its new office buildings and research laboratories with type 202. Long lists can be made of employment during this period in types 201 and 202 in household, restaurant and



Cherry-Burrell Corporation, Chicago, Ill., leading manufacturer of dairy and food processing equipment has produced several units in Cr-Ni-Mn steel on a trial basis. The composition used contained 17% Cr, 4% Ni, 10% Mn, similar to Type 202.

Two of the bowls for small bottle fillers shown above were produced in 1953 and have been in service since that time without any indication of reduced service life as compared to Type 302.

These test parts and other items of dairy equipment made by various manufacturers indicate that Types 201 and 202 stainless steel can be used alternately with Types 301 and 302 for non-contact surfaces and that, perhaps, they may prove equally satisfactory for contact surfaces as in the two cases above.

Bradley washfountain bowls are drawn to size in a single draw from stock .095" thick. Blanks for the 36" bowls are 48" in diameter; the larger bowls are drawn from 66" blanks. Other operations include curling the edge of the bowls with a wiping die, machining the curled edge and polishing.

In tests made on Type 202 no production changes were required nor were any significant differences between Type 202 and 302 observed.

miscellaneous applications fields. Some of these are: coffee makers, pots and pans, teakettles, flatware, sinks, refrigerators, wall panels, range hoods and other parts, food mixers, steam irons, washing machines, wash fountains, dispensary cabinets, hose clamps, electric pole line hardware, and capacitor cases. Several of these applications are pictured in the illustrations accompanying this article.

The Committee of Stainless Steel Producers of AISI (1)* has reported as follows: *There has not been a single unsuccessful application of Cr-Ni-Mn steels discovered to date and many estimates place eventual use of these steels at 75 to 80 per cent of the applications now enjoyed by types 301 and 302 . . . The best estimate of the value of these steels can perhaps be found in the belief of a number of persons that production of types 301 and 302 could, if necessary, be stopped without any major effects.*

Gradually during the period 1950-1957, it was realized that, like so many other recently developed materials—nylon, synthetic rubber, war emergency alloy steels—the 200 series stainless steels are not just substitutes designed to meet an immediate emergency, but dependable and worthwhile products, in their own right. The 200 series steels have now been accepted by many as a new family of stainless steels with a real future of their own—nickel shortage or no.

Since the 1957 recession, it would almost be an understatement to say that nickel has been readily available—in truth it has been in recent months, for the first time in history, in great over supply. Government controls and stockpiling have been curtailed. For the development of these particular steels, unfortunately, the real push was lost just as they were about to go over the top. Nevertheless, both their instrinsic value of-and-by themselves and the fact that in any war emergency nickel will again be in short supply, should be recognized for evaluating their merit.

Since the Sputnik era, it has been quite popular to keep close tab on technical progress behind the iron curtain. It is most interesting then to note the extreme interest in manganese-bearing stainless steels in Soviet Russia. (2) There is reportedly no shortage of nickel in that country and yet leading Soviet metallurgists foresee a future shortage and look to these very alloys as a relief. There is thus considerable research going on into the use of these steels, and an electrolytic manganese plant has been constructed in Agledzia, Georgia, SSR.

CHEMICAL AND PHYSICAL PROPERTIES

The tables below list some corresponding physical and chemical properties of 201 and 202 compared with 301 and 302 types:

A cursory glance at the tables reveals that

Malt cups and similar pieces shown here are being produced by Bloomfield Industries, Inc., Chicago, Illinois, using Type 202 stainless steel. These cups were previously made with Type 302 and the change to the lower nickel alloy did not require any changes in production.

Throughout all operations including drawing, curling, polishing and spot welding, there were no noticeable differences between Type 202 and Type 302.

* Numbers in parenthesis apply to references at the end of this article.

Chemical Compositions (3)

	201	301	202	302
Cr	16.0-18.0	16.0-18.0	17.0-19.0	17.0-19.0
Ni	3.5- 5.5	6.0- 8.0	4.0- 6.0	8.0-10.0
C	.15 max	.15 max	.15 max	.15 max
Mn	5.5- 7.5	2.00 max	7.5-10.0	2.00 max
Si	1.0 max	1.0 max	1.0 max	1.0 max
N	.25 max	-	.25 max	-

Physical Properties (1)*

	201	301	202	302
Yield strength (.2%) psi	55,000	40,000	55,000	40,000
Ultimate Tensile Strength, psi	115,000	110,000	105,000	70,000
Per cent Elongation (2")	55	60	55	50
Hardness, Rockwell-B	90	85	90	85

* Sheet material, 2B finish.

the saving in nickel in 200 over corresponding 300 grades is about 50%. Physical properties, however, are comparable but not identical. A significant difference is apparent in yield strength—the designer's principle parameter. Here it should be noted that in many applications, especially where weight or space are important, the higher strength-weight ratio of the 200 to the corresponding 300 series steel would result in a distinct advantage in service. In particular, the yield strength of type 202 in the annealed condition is over 35 per cent greater than that of 302. Where such a strength edge obtains both economy and weight advantages result.

At elevated temperatures, the creep behavior of types 201 and 202 has proved superior to the 301 and 302 types, and compares favorably with stabilized type 347. (4) The advantage of 200 over 300 types is even more marked in short time tests at elevated temperatures. At 1200-1400°F, the yield strengths of types 201 and 202 are almost double those of type 302. (5)

PRODUCTION PRACTICES AND FABRICATION

Although yield and ultimate tensile strengths of the 200 grades are higher, this only mildly affects production practices. Work-hardening characteristics are similar. Generally, at the same strength levels Cr-Ni-Mn steels are more ductile than the straight Cr-Ni steels. Draw-bench, roll-forming, and brake-bending operations require no change in techniques. Shallow and deep-drawings operations require some change in technique to prevent wrinkling. Longer hold-down period is recommended. Draws up to 36" in diameter by 10" deep are possible. (1) (6)

Types 201 and 202 may be welded by all arc-welding, spot, seam, and projection-welding procedures suitable to 301 and 302 types. (7)

The 200 series have somewhat lower resistance to scaling over 1600°F. For this reason anealing temperatures are usually reduced 50-100° or so to promote lighter scale.

Types 201 and 202 are easier and more economical to finish by grit polishing, buffing, and flash chrome-plating. The same procedures may be employed as are used for types 301 and 302. The final surface appearances are the same. (5)

CORROSION RESISTANCE

Corrosion resistance—after all the most important

Tuttle & Kift, Inc., Chicago, Illinois, manufacturer of electric range burners has made extensive tests on Type 201 stainless steel as an alternate to Type 301 for burner rings and supports.

Burner rings are produced by initial blanking and drawing as shown here followed by redraw, blanking and polishing operations. No. 2 finish strip is used for the rings; .020" strip is used for 6" diameter rings, .023" strip for 8" diameter rings.

factor for a "stainless" material—shows a favorable comparison all around. Types 201 and 202 have excellent atmospheric corrosion rates as evidenced by their 15-year use by the Budd Company, of Philadelphia, in railroad cars and truck trailers. The International Nickel Company has conducted a 14-year industrial atmosphere corrosion test successfully, showing no difference between the types. The 200 types have proved satisfactory in every application in which they have been employed, for cooking utensils, sinks, and restaurant equipment. (6)

It is noteworthy that much of these sanitary service usages are applicable to shipboard use.

Most relevant to the interests of present readers may be the salt vapor tests also conducted by INCO. In these tests Cr-Ni-Mn samples were comparable in appearance to 18-8 Cr-Ni samples exposed to a marine atmosphere for 9 years! There was no decrease in tensile strength, yield strength or ductility. The samples were not cleaned during the entire period but the oxide coating which developed could be removed without difficulty. (1)

Corrosion resistance to certain chloride compounds is poorer in the 200 than in the 300 series. In these instances, however, use of 301 and 302 has been marginal. (6) Field tests in formaldehyde solution indicate that 202 is as resistant to corrosion as type 304. (4)

Actual corrosion data on 201 compared with 301 is given below for typical examples. (Data in inches penetration per month.) (7)

	201	301
Boiling 65% HNO_3	.0017	.0015
Boiling Cu_2So_4, H_2So_4	.00003	.00002
5% Lactic Acid	.00001	.00001
10% H_3PO_4	.00001	.00001
60% Acetic Acid	.00001	.00001

Other data compiled by the Committee of Stainless Steel Producers of the AISI show a comparison of types 202 and 304 subjected to various reagents, including 2 per cent HC 1. Corrosion rates are shown to be similar. This Committee has made the statement, "These steels (201 and 202) can replace types 301 and 302 in any application where the latter steels are being properly used. While they are not generally recommended as replacements for type 304, there may be instances where steel specifications are 'overalloyed.' In these cases, the low nickel, austenitic types may be able to perform satisfactorily. Typical are certain dairy items which were previously specified in type 304 and subsequently changed to type 302." (7) Many shipboard uses of stainless steel in commissary and sanitary spaces fall precisely in this category.

The dairy industry has studied but not yet approved type 202 for contact with milk products. They, too, have specification inertia to overcome.

SCRAP HANDLING

Scrap segregation of 201 and 202 steels should be practiced in order to afford the maxium opportunity for alloy recovery. Preferred practice is to melt down the initial charge of carbon and low-manganese scrap (300 and 400-series). After an oxygen blow 200 series scrap is back-charged. By this method it is possible to recover up to 85 per cent of the manganese and 90 per cent of the chromium from 201 and 202 scrap.

AVAILABILITY

Types 201 and 202 are available in commercial quantities in sheets and shapes, some warehoused, others on special order. Some types, such as Allegheny-Ludlum 204—a replacement for type 304—are available only in experimental quantities. Some experimental 200 series steels also contain Nitrogen as an austenitizing agent. None of the latter are yet commercially available. Most of the companies approached in writing this article indicated that they are planning to warehouse items as the commercial demand for these new products expands. Crucible Steel Company, for example, plans to stock 201 and 202 (their brand name "Rezistal 201 and 202") in gauges USS 12-24; widths 36 and 48 inches, lengths 96 and 120 inches; finishes 2B, polish 3 and 4.

The actual gain in consumption of the 200 steels has been encouraging although not by any means skyrocketing. Production of these steels has risen as follows as a percentage of austenitic stainless steels:

First quarter of	1956	1957	1958
	2.8%	3.2%	4.7%

PRICE

The price differential between 200 and 300-series steels is not now very great. This is partially due to the present small demand, brought by the limited knowledge of the existence of these new stainless steels on the part of many Government and industrial users, and the natural caution of engineers to alter specifications. As the demand grows, the price should drop to about midway between the straight chromium (400) and chrome-nickel (300) types. At the present time, the price differential is about 2c per pound, as is shown by the table below.

Type—Price in Cents/lb., Subject to Extras (14)

	Sheets	Strips	Shapes
201	48.50	36.00	42.00
202	49.25	39.00	43.00
301	51.25	37.25	44.25
302	52.00	40.50	45.00
304	55.00	44.25	47.75
430	40.75	32.00	34.25

EXISTING SPECIFICATIONS

The Department of the Navy uses many specifications for stainless steels. The equivalent chemical compositions of Government Specifications and AISI

types are to be found in "Supply and Logistics Handbook," Standardization H-1A, published by Assistant Secretary of Defense (S&L).

Although the correlation is not exact, some Navy Department, Military, Federal, and ASTM specifications are listed below. The table shows only the specifications for flat products; other specifications cover structural shapes, wire, piping and tubing.

Type 301

N. D. 47S30, C1.2	Plate,	Sheet	and	Strip	(Clad)
Fed. QQ-S-682, FS301	"	"	"	"	
Mil - S-505 9	"	"	"	"	
Fed. QQ-S-766 (Ships), FS301	"	"	"	"	
ASTM A-167-54, Gr. 1	"	"	"	"	

Type 302

MIL-S-5059A (ASG) Type 302	Plate,	Sheet	and	Strip	
Fed. QQ-S-682, FS302	"	"	"	"	
Fed. QQ-S-766 (Ships) FS302	"	"	"	"	
ASTM A-167, Gr. 2	"	"	"	"	
MIL-S-854 (Ships) C1.2	"	"	"	"	and Shapes

Type 304

Fed. QQ-S-682, FS304	Plate,	Sheet	and	Strip	
MIL - S-854, (Ships) C1.1	"	"	"	"	
Fed. QQ-S-766 (Ships) FS304	"	"	"	"	
N. D. 47-S-30, C1.1	"	"	"	"	(Clad)
N. D. 47-S-20, C1.1	"	"	"	"	
ASTM A-167-54, Gr. 2	"	"	"	"	

Federal Specification QQ-S-00766b (plates, sheets and strips) has been revised to include both 201 and 202 types. ASTM specification A-412-57T (tentative) likewise covers AISI types 201 and 202. No "MIL" or Navy Department specifications cover these types.

SPECIFIC EXAMPLES FOR SHIPBOARD APPLICATION

Use of CRES aboard ship is widespread. Primarily, it is used for its corrosion-resistant property, but, occasionally, it is used for other specific properties, such as high temperature strength, or low magnetic permeability. Austenitic stainless steel, principally of the 302 and 304 varieties, is specified for use aboard Navy vessels for many purposes. *General Specifications for Ships,* Section S1-0, lists these non-critical uses: cold storage gratings, coamings for shower spaces, sheathing for cold storage, dresser tops, drain boards, shelving for meat stowage, mess counter tops, sinks for photographic laboratories, steam tables. Commercial vessels employ the same materials in similar fashions and have additional peculiar applications. Extensive use is made of stainless steels in many living quarters and in particular liquid and cold storage cargo spaces.

When considering the substitution of 200 for 300 series steels the most obvious examples are those originally specified for its room temperature resistance to mild corrosion, as in living, commissary, and sanitary spaces. In some instances—considered the best examples—direct substitution of a 202 steel for a specification corresponding 304 type is recommended. It is believed that, in many cases, a type 302 would have served as well, anyhow. It should be remembered that 301, 302, and 304 should never be recommended where the chloride ion is present in excess, (e.g. in direct sea water submersion), or where lack of oxygen exists, (e.g. under gaskets). However, where considered absolutely essential to have the low carbon properties of the 304 type specified rather than 302, then the new 204 steel should be considered as a substitute.

Just for example each ship of a recent class of Navy LST's employed some 27,200 pounds of types 302 and 304 stainless steels readily amenable to substitution by 201 and 202. Included in this sum are cold storage sheathing and doors, galley sinks, joiner bulkheads, dresser tops, cabinets, coamings and innumerable minor installations.

It is not the purpose of the present article to delineate in great detail the many instances where replacements can readily be made. Actually, the author has reviewed the Navy's general specifications and the complete compilation of Hull Type and Standard drawings currently in use for this purpose. In the resulting Value Engineering Project an extensive list by piece and part was compiled. With proper selection and enlightened designing, the opportunity is readily available for wider and wider use of these new materials on both Navy and merchant ships. This use in turn will gain greater acceptance in other areas, a consequent reduction in price, and thus an even better competitive position.

The bibliography at the end of this article includes most of the known technology in the production, fabrication, and application of the 200-series steels.

CONCLUSION

A case has been presented for the use of a recently developed series of stainless steels. Required for its wider application in the marine industry is familiarization on the part of design agents and specification engineers with its advantages and limitations, its availability and price, and every other facet of the series in question.

A quite similar case could be made for many other recently developed construction materials; many of these are now shelf items, often, as in this case, already accepted for use in other industries. Certainly many such materials as organic adhesives, plastic piping, prestressed concrete, automatic fastenings, silicone lubricants, banded protective coatings, and glass reinforced resins belong in this category. All of these are exciting new materials worthy of critical evaluation for marine use. Many are now getting just this attention already. Of course, reckless adoption of new materials for the sake of change alone is unwarranted, but enlightened de-

sign can make advantageous use of these and hosts of other presently available products. By such enlightened designing is true progress made.

ACKNOWLEDGMENTS

The author is indebted to the following organizations for advice and information:

American Iron and Steel Institute, New York
Allegheny-Ludlum Steel Corporation, Pittsburgh
Crucible Steel Company of America, Pittsburgh
Republic Steel Company, Cleveland
Electro Metallurgical Company, New York

All the photographs for this article are used with kind permission of the Committee of Stainless Steel Producers, American Iron and Steel Instiute.

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APPENDIX

"DEPARTMENT OF DEFENSE
OFFICE OF THE SECRETARY

June 14, 1956

Dear Mr. Fairless:

The continued shortage of nickel to meet current needs of our industry gives me particular concern in view of the large quantities of this metal that are currently required in the production of aircraft engines and other military items. The prospect is that even larger quantities of nickel will be required for the new designs of military equipment that will come into production in the years ahead of us.

In view of this prospect, the contribution of the American Iron & Steel Institute in bringing about formal recognition of the manganese-nitrogen stainless steels (AISI 201 and 202) is deeply gratifying to the Department of Defense as it must also be to you. This is certainly a major national contribution to effective use of materials, whether for purposes of war or peace.

I urge that every effort be made to see that these two new stainless steels become the forerunners of a broader, more inclusive series. If it is practicable to develop additional low-nickel counterpart steels, in the 200-series, to exploit fully the applicability of the manganese-nitrogen principle, the development and use of such alternates would appear to be very much in the national interest at this time.

Sincerely yours,
(Signed) Frank D. Newbury
FRANK D. NEWBURY

Mr. Benjamin F. Fairless
President, American Iron &
Steel Institute
150 E. 42nd Street
New York 17, New York."

"AMERICAN IRON AND STEEL INSTITUTE
150 East Forty-Second Street
New York 17, N.Y.

July 2, 1956

Mr. Frank D. Newbury
Assistant Secretary of Defense
Applications Engineering
Washington 25, D.C.
Dear Mr. Newbury:

It is gratifying to know that the efforts of the stainless steel industry to conserve nickel through the development of the chromium-nickel-manganese-nitrogen stainless steels (200 series) has met with your attention and approval.

You will be pleased to know, I am sure, that considerable promotional effort is being placed behind the wider adoption of these grades both by the producers themselves and our Committee of Stainless Steel Producers.

In addition, various of the stainless steel producers are now making low nickel manganese- nitrogen steels in other variations equivalent to Type 304 (lower carbon), Type 303 (free machining), and so on. As these are made in greater quantity they too, no doubt, will warrant wider recognition.

The next major contribution to the use of the lower nickel stainless steels must come from your own department. Until your people change their material specifications to permit the use of these steels in appropriate applications, your suppliers and others who copy your lead in their own specifications will, necessarily, have to use the higher nickel grades.

This action, in fact, could be taken with respect to other government specifications under which high nickel steels are required. There are many instances where such steels are used when the end-use performance does not require them.

Our technical committees would welcome the opportunity to assist your department in devising alternate steels of low nickel content, or even nickel free steels, to do those jobs without sacrificing safety or impairing performance.

Very sincerely yours,

/s/ B. F. FAIRLESS
President"

A novel hull form for boats has been developed and tested by Higgins, Inc. The builder has applied for a patent for the unique underwater hull surface, which has been named the Polyhedral design. In this design the surface of the shell plating from the keel to the chine in section has an exaggerated clinker, or sawtooth appearance. The longitudinal grooves, or steps, so formed tend to prevent the lateral flow of water from keel to chine, restraining it to flow in a longitudinal direction. This acts to prevent slamming of the hull against waves by providing a cushioning effect. The result of the greatly reduced impact is a smooth ride and excellent seaworthiness. Two Solar gas turbines rated at 500 hp each power the boat at over 40 mph. Airplane type reclining seats in the cabin are provided for 30 passengers.

—from MARITIME REPORTER
February, 1959

The Argonne Low Power Reactor, a 300 KW electrical output, boiling water reactor, has undergone a series of tests since criticality last August. The reactor is designed to produce heat and electric power at remote locations. About 85% of the steam produced at 300 psig is used to drive the generator turbine. The remainder is available for space heating. A unique feature is the thermal circulation system within the reactor vessel which causes about 130 lbs. of water to be circulated through the reactor core for every pound of steam produced. After reaching full power in late October, the reactor was shut down and restarted in eight hours, at peak xenon poisoning, to demonstrate its capacity to override peak xenon concentration. No single component of the plant weighs more than 10 tons nor has dimensions greater than 20 x 7 x 9 ft. The core is designed to operate continuously for three years without reloading.

—from NUCLEONICS
February, 1959

Official U. S. Navy Photograph

Somers **(DD 947) and** ***Dewey*** **(DLG 14) contrast two modern types as they near completion.**

"THE SHIPBUILDER AND MARINE ENGINE-BUILDER"

TECHNICAL PROGRESS IN SHIPBUILDING DURING 1958

ACKNOWLEDGEMENT

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In discussing the technical progress during the past year, reference will be made to much of the work published by the technical institutions, but it is fitting, at the outset, to make a few general comments. In this country, the British Shipbuilding Research Association is one of the major organizations initiating and directing research and we cannot do better than refer to the general program of research of this organization in order to get some idea of what is being done at the present time. The work of the Association over the past seven or eight years was described by the Director of Research, the late Mr. S. Livingston Smith, in a paper during the year.[1] This gave an indication of the organization of the Association and described how an endeavor is made to investigate the problems which are of immediate importance to the shipbuilding industry. To date a very wide range of work has been dealt with by the Association, and it is of great interest to note that similar organizations have been set up and developed in other countries. At the present time the British Shipbuilding Research Association still have in hand, in the resistance field, the work of merchant ship forms whose block coefficients lie in the range 0.65 to 0.80, and it is hoped, eventually, that it will be possible to arrive at a simple means for both drawing the lines plan and determining the resistance for any form in the family.

Another subject still being pursued by the British Shipbuilding Research Association is the determination of stresses in ships' structures, and to this end several ships have been fitted with autographic and statistical strain gauges. Work is also being carried out on the determination of bending moments on ship models among waves, in the experimental tank of Messrs. Vickers-Armstrongs (Shipbuilders), Ltd., at St. Albans.

Ship vibration is still a subject which is receiving the attention of the British Shipbuilding Research Association, and at Bristol University panels of ships' plating are being investigated. At the Ship Division of the National Physical Laboratory shallow-water effects are being studied. The work in the Department of Naval Architecture at King's College on the vibration of free-free beams continues, and investigations have been made for the British Shipbuilding Research Association on the influence of three dimensional flow on the added virtual mass of prismatic beams of circular, elliptical and midship-shape cross section.

The British Shipbuilding Research Association have also completed a program of exciter tests in three modern trawlers of different sizes. Vibration measurements on board ship are still being made as opportunity arises.

A further problem being examined by the British Shipbuilding Research Association is the ventilation

of cargo spaces. This work is being carried out at the Royal College of Science and Technology, Glasgow.

Progress is being made with the new research facilities for the Ship Division of the National Physical Laboratory. Mention was made of this work last year and now it can be stated that the new equipment at Feltham will be in operation in the not too distant future.

The foregoing brief account gives some idea of the scope of the research work being carried out in this country. More detailed acounts of some of this work will be found in the following sections, and also indication will be given of work in other countries, particularly the United States of America.

STRUCTURAL STRENGTH AND WELDING

In this field, three very valuable American papers have been published during the year.

The first of these papers [2] is concerned with the various full-scale strength experiments which have been carried out in the past ten years or so. The author examines some 19 different sets of experiments and some of the conclusions are extremely interesting. For instance, it appears to be firmly established that, for all practical purposes, the longitudinal structure of a ship obeys the ordinary beam theory with sufficient accuracy. Ships which have long deckhouses, however, do not obey simple beam theory, and, as is now well known, the stresses in such deckhouses are not proportional to the distance from the neutral axis. Riveted and welded ships behave in much the same way, insofar as deflection and stresses are concerned, but the welded ship is more liable to panel buckling than the riveted ship because of greater unfairness of plating. Longitudinal framing is to be recommended to provide a greater resistance to buckling, and the limiting bending moment for a longitudinally framed ship is given as:— Bending moment = panel strength × section modulus.

The second of the papers [3] referred to is concerned with model experiments on the strength of ships among waves. This is a relatively new subject of research and one which shows increasing possibilities. The present research was carried out at the Mejiro Experimental Towing Tank in Tokyo and the results of tests on two brass models—one with U sections forward and the other with V sections forward—are recorded. The models had the same principal dimensions. The pressures induced by the waves were measured, as were the hull stresses, at various points in the models. It was apparent from the results that slamming occurred when synchronism between the pitching period and the period of encounter of the waves was approached and that the V form was superior to the U form from the point of view of the prevention of slamming. The stresses set up in the hull were independent of ship speed until synchronism was approached between the period of encounter and the pitching period. Then the stress gradually rose. Added to this was a considerable instantaneous increase in stress due to slamming. This slamming stress had its greatest value some distance forward of amidships, and not at amidships, as might be expected. From the results of the experiments it can be concluded that the ship with the least flat bottom in the bow area, associated with a good raked stem and a gentle turn of bilge, would be the most satisfactory type of ship to reduce slamming.

The third paper [4] deals with a design method for the longitudinal material of the structure of a ship. The information upon which the method is based is taken from the rules of the American Bureau of Shipping. It is assumed that these rules represent the most comprehensive data available that are reasonably representative of the lower limits of merchant ships structural adequacy. The various items of the longitudinal structure of the ship are dealt with in turn, and design methods and formulae given for obtaining the scantlings of these parts. The technique described is very flexible owing to its reliance on basic concepts of structural mechanics, and it should enable the determination of the scantlings of ships of greater or smaller size and different proportions from those given in the rules. Also it should be possible to deal with quite different internal arrangements, structural arrangements and framing systems.

A most useful paper by Mr. J. M. Murray, Chief Ship Surveyor of Lloyd's Register of Shipping,[5] was given recently. This is concerned with the longitudinal strength of tankers and is most helpful to naval architects in that it gives an insight into the methods adopted by Lloyd's Register of Shipping in dealing with this type of ship. The total longitudinal bending moment is divided into its two component parts, *viz.*, the still-water bending moment and the wave bending moment. The former is calculated according to methods laid down in the author's earlier work. The wave bending moment is the more difficult to assess, as this is dependent upon the height and length of the waves assumed. It will be of interest to those concerned with the subject of strength calculations that the standard wave now adopted by Lloyd's Register of Shipping for such calculations has a length equal to the length of the ship and a height given by $h = 1.1\sqrt{L}$. This is a departure from the well-known L/20 wave, and it will be seen that it gives more severe conditions for the shorter ship and less severe conditions for the longer ship. Thus, at L = 400 ft. the wave height would be 22 ft. instead of 20 ft. and at L = 900 ft. the height would be 33 ft. instead of 45 ft. The method of obtaining the section modulus in tankers permits modifications, from a minimum value for variations in the bending moment, and it is clear that each ship can be treated on its merits, *i.e.*, various design features can be taken account of in assessing the scantlings. This procedure is one to

be commended and should lead to realistic and economic structural design.

The subject of brittle fracture in welded ships is one to which attention has still to be given and the shipbuilding industry should welcome a further contribution on this matter by Mr. J. Hodgson and Mr. G. M. Boyd.[6] In this most recent paper an analysis of the casualties due to brittle fracture is made and some general conclusions are drawn which should act as a guide for the future. In this most intriguing problem there are two important factors; first of all the design of the structure, which governs the general stress level, and the stress concentrations which are likely to take place; and, secondly, the material itself. It would appear from the general conclusions made in this study that it is not practicable to reduce present design stress levels sufficiently to provide adequate safety by this means alone. It is admitted that stress concentrations cannot altogether be avoided and, consequently, under the operational and structural conditions imposed, prevention of serious fracture must depend primarily upon the resistance of the steel, as built into the structure, to the propagation of fractures. In view of this a minimum measure of notch toughness is required, and Lloyd's Register of Shipping have now incorporated in their Rules the requirement for a Charpy V-notch impact test. The minimum energy required from this test is 35 ft. lb. and 30 per cent minimum fibrous. These figures are at a temperature of 0°C. The authors of this paper, however, stress the importance of ensuring the highest standards of structural design and welding.

A paper has been read during the year on the subject of records of fractures in welded ships and their courses.[7] One of the most interesting items referred to in this paper is the system adopted by Lloyd's Register of Shipping for recording and classifying reports of damage sent in by Surveyors all over the world. The system was initiated in 1946 and has developed since then, and many successful investigations have been carried out with its aid. Some of the places where fractures occur are bilge keels, intersecting members, cracks at rivet holes, run off plates, T-junctions, and welded accessories. Many of these troubles are of a relatively minor nature, but are, nevertheless, potentially dangerous. It is interesting to note that many shipbuilders have visited Lloyd's Record Department and have with the owners' consent made notes on fractures occurring in ships built in their own yards. This procedure can help in preventing constructional faults being repeated in future ships and will lead to improvements in ship construction. It is suggested that more shipbuilders may with advantage avail themselves of the facilities afforded to them in this way by Lloyd's Register of Shipping.

Turning to theoretical studies of the strength of ships, the problem of transverse strength has once again been discussed.[8] In this analysis the slope deflection method is used rather than the Hardy Cross method, which has become popular in recent years. The author claims that the former method is more suitable for taking into account the spacing of transverse bulkheads and the influence of longitudinal girders on transverse strength. The method is used to develop the equations for the bending moments in the transverse framing ring of a single-deck ship. The equations take into account variations of draft, internal loading and distribution of internal supports. The effect of heeling has also been considered. The advantage of the method of analysis proposed is that a given ship can be taken, and the end moments obtained, merely by inserting, in the equations, the tabulated fixed-end moments for the various members under any desired condition of internal and external loading. The transverse strength calculation thus becomes a routine. It is useful for examining the effects of abnormally long holds or hatches, of the adoption of cantilever construction or of an unusual transverse distribution of loading.

Interest still continues in the field of aluminum alloys in ship construction, and, in fact, at the present time some large structures are being considered. Naval architects will, therefore, welcome some results of tests on light alloy sections carried out by the British Shipbuilding Research Association and published in November last.[9] This work was undertaken to expedite the development of sections for marine purposes. The sections concerned were 9 in. channels and bulb angles, each in a range of web thicknesses. Bars were extruded from both the shipbuilding alloys N6 and H30, and these were tested for their suitability in shipbuilding. The structural tests were carried out in the well known testing machine at Glengarnock, and shipyard forming tests were carried out at the works of Messrs. Alexander Stephen & Sons, Ltd. The structural tests showed that, within the elastic range, stress and deflection were proportional to the section modulus and moment of inertia respectively. The shipyard tests showed that all the sections were suitable for punching, joggling and bevelling.

RESISTANCE AND PROPULSION OF SHIPS

At the beginning of this section on resistance and propulsion, it is fitting to make some reference to developments in research as they have been recorded in published literature. A paper on developments at the David Taylor Model Basin was read to the American Society of Naval Architects and Marine Engineers in June last.[10] It is obviously not possible to describe in any detail the various items of equipment discussed in this paper, but it may serve to indicate the scope of these facilities if we enumerate them briefly. In the field of ship motions, for example, a planar-motion mechanism provides hydrodynamic coefficients for the differential equations of motion. Also in this field there is a heaving towing point mechanism which simulates ship pitching for bodies towed over the stern. With regard to tests in waves, the David Taylor Model Basin has a

pneumatic wave maker which can generate a programmed frequency spectrum. A flutter dynamometer has been developed which permits investigations of control surfaces such as rudders in water. In the propeller field, a large variable-pressure water tunnel provides for the testing of contra-rotating propellers. The addition of sea keeping and rotating arm tanks opens up a further field in naval architecture research. With reference to the ordinary towing facilities it may be noted that there is a 6-knot carriage capable of towing a 57-ft. model, and at the other end of the scale there is a towing carriage capable of operating at 60 knots in a tank nearly 3,000 ft. long. With all this equipment is associated the latest developments in instrumentation and the use of analog and digital computers plays a large part in the research work envisaged with this apparatus. Undoubtedly, with all these facilities available useful results should be forthcoming and we can expect, in the future, that many of the hitherto abstruse hydrodynamical problems will be solved satisfactorily.

Turning now to more specific problems, we would refer to a paper read by Mr. A. J. W. Lap.[11] This paper deals with skin-friction extrapolation and is particularly concerned with the three dimensional problem. The three dimensional aspect of skin friction has been much to the fore in recent years and this present work makes a further important contribution to its study. It will be remembered that the International Towing Tank Conference adopted a skin-friction line in 1957, known as the I.T.T.C.-1957 line, and the present author regards this as a suitable basis for current three dimensional extrapolation methods. It is considered that the magnitude of the dimensional form factors is mainly determined by the form parameters C_v and L/H. The value of B/H is not of any great importance in determining the form effect. One of the troublesome factors in extrapolation is the scale effect of appendages and it is, therefore, of interest to note that this effect is automatically taken into account if three dimensional extrapolation methods are employed. Another important fact which emerges is that the ratio of the frictional resistance to the total resistance is much greater than is generally supposed. Also general wave resistance calculations are in better agreement with experimental results if the frictional resistance is calculated three dimensionally. These can be clearly seen by the diagram in Figure 1, which is reproduced from the paper.

With regard to motion in restricted waterways, it should be noted that the form effect on frictional resistance is dependent on the relative water depth and water width as well as C_v and L/H. Roughness allowances for ships are dependent on the form factor and, therefore, on the ship form and the dimensions of the waterway.

The study of the boundary layer of actual ships is being continued and we note further work on this subject carried out by the National Physical

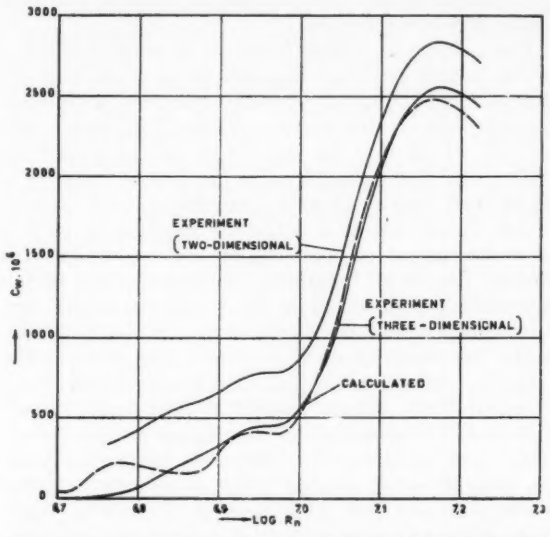

Figure 1. Comparison between Calculated Wave Resistance and Experiment Results.

Laboratory.[12] This work consisted of a series of experiments to determine the relation between the boundary layer velocity, the distance from the surface of the body and the shearing stress τ_o at the wall. The first experiments were on pontoons at two speeds and the law deduced for these was

$$\frac{u}{u_*}=4.98\log\frac{u_* y}{v}+6.54.$$

In this formula u is the velocity at distance y from the surface, while

$$u_*=\sqrt{\frac{\tau_0}{\rho}}$$

The formula agrees very closely with that obtained by Ludweig and Tillmann but differs considerably from the more commonly accepted Nikuradse value. In addition to the work on pontoons, velocity measurements were also made on two ships. The first of these was a trawler on which two traverses were made at approximately 0.6 length from the bow. The traverse taken on the flat bottom gave a typical velocity distribution curve but that carried out in the side position showed marked fluctuations in the value of $\frac{u}{U}$ with increasing y. The second ship was an ore-carrier and traverses were made at six positions around a girth at 0.3 of the ship's length from the bow. Considerable variations were experienced of both the boundary layer width and the free stream velocity at the edge of the boundary layer. Attempts were made by using the pontoon results and Ludweig Tillmann's formula to find C_f (the local specific resistance coefficient) at each of the positions but the disagreement between the two methods throws some doubt on the validity of their use in three dimensional

flow. Figure 2 gives an indication of the velocity distribution in the boundary layer in the trawler *Sir William Hardy* and also shows how fouling affects the velocity distribution.

In previous years we have made reference to work on "Victory" ships and their models and we have to record a further contribution on this subject.[13] This latest work is concerned with wake measurements and their analysis. Results of wake measurements behind seven "Victory" ships on scales from $\frac{1}{50}$ to $\frac{1}{18}$ and on the model boat *D.C. Endert Jnr.* on $\frac{1}{6}$ scale were analyzed. The wake fractions, when plotted to a base of Reynolds Number, showed a steady decline with increasing Reynolds Number. Thus at R.N. $= 10^6$ the value of the wake fraction would be about 0.46 and at R.N. $= 10^8$ it would be about 0.31. This clearly demonstrates the danger of accepting, without correction, wake fractions from model experiments for use on the full size ship. A means of extrapolating wake fractions is suggested in this paper. This may be stated as follows:—If the measured mean-wake fractions and the theoretically derived friction-wake fractions of the propeller disc, behind a flat plate, are plotted to a base of Reynolds Number they can be related to each other by a frictional-wake form factor. This factor can be derived from the frictional resistance form factor so that, if the latter is known, it should not be impossible to forecast the wake for the full-size ship. With regard to wake distribution over the screw disc, it was found that the difference between the maximum and minimum in the circumferential distribution of the wake was unaffected by Reynolds Number. With increasing Reynolds Number, however, the area of high-wake values decreases in width in consequence of the smaller boundary-layer thickness.

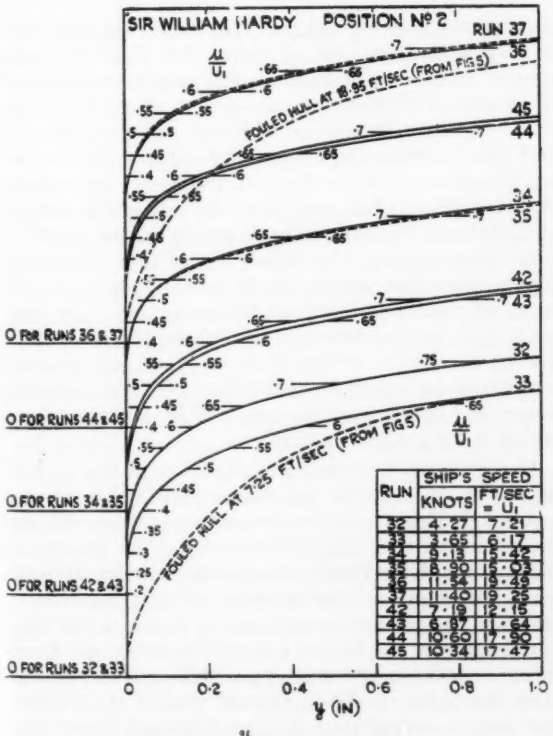

Figure 2. $\frac{u}{U_1}$ for Clean Hull.

The results of a series of full-scale trials on inland and shallow-draft vessels have been made available during the year.[14] These trials were carried out by the Rivers Steam Navigation Co., Ltd., and the India General Navigation and Railway Co., Ltd., in conjunction with Messrs. William Denny & Brothers, Ltd., of Dumbarton. The main objects of the trials were to compare the full form for non-powered flats in deep and shallow water, to compare, when running free and under a variety of loads, an open screw and a Kort nozzled steam tug and also to compare abreast, astern and push towing for a variety of vessels in deep and shallow water. The results of these rather extensive trials showed that the improved and finer lines of the new type "Deep Sea flats" afforded easier propulsion than the older type "River Flat" for equivalent displacement. It was also demonstrated that for the loads imposed, and speeds at which these trials were conducted, the performance of the open-water screw tug was, in all cases, excepting the bollard pull test, better than the Kort nozzle tug. It was further shown that push towing was the most efficient with astern and abreast following in this order in all depths of water.

The use of hydrofoils for high-speed craft has been studied for many years now and a further useful contribution to the subject has been made by Mr. P. R. Crewe.[15] The paper to which we make reference is partly historical in that it traces the development of this interesting type of craft; it also gives some account of the future prospects of such vessels. Hydrofoil craft have advantages over the conventional high-speed planing craft, the main one being that they have lower resistance. It is claimed that they are more comfortable in rough seas and that they can operate satisfactorily in waves up to 6 ft. in height, depending upon the size of the boat. The advocates of this type of craft consider that a new field of marine craft performance is opened up for ships which can travel at 70 knots with a lift-drag ratio of 15 and a propulsive efficiency of 0.65. The detailed considerations in the design of such craft are cavitation, static and dynamic stability and structural requirements. It is of interest to note that special purpose designs are now possible which have speeds up to 60 knots, but at the present the disposable load is small.

We conclude this section by making reference to an interesting series of tests on the roughness of propellers, described by Mr. J. M. Ferguson.[16] This work was carried out in the experimental tank of Messrs. John Brown & Co. (Clydebank), Ltd., and was done at the request of the New Zealand Shipping Co., Ltd. The investigation was made because the growth of small shells on the blades of propellers while ships were in New Zealand waters and harbors appeared to cause a marked deterioration in the performance of the ships during the homeward voyage. To study the problem it was decided to carry out open-water tests on a 24 in. diameter model of the propeller fitted to the *Otaki* with varying degrees of roughness on the blades. In addition to the uncoated propeller eight tests were made with different roughnesses, the grain size being up to 0.028 in. for the roughest propeller. The model itself represented a full-size propeller the characteristics for which are given in Table I.

TABLE I

Diameter	19 ft. 3 in.
Mean face pitch	15.5 ft.
Pitch ratio	0.808
Number of blades	4
B.A.R.	0.652
Root thickness÷diameter	0.0418
Boss diameter	3 ft. 9 in.
Rake	15 deg.

The main points from the results of the tests are:

(1) Loss of efficiency increases with grain size, at first very rapidly, but it would appear that there will not be much further loss of efficiency for grain sizes greater than 0.028 in.

(2) The influence of roughness is greater at low slips than at high slips as might well be expected, since it is well known that friction forces have a greater influence on propeller performance at low slips.

To quote some of the results, the maximum loss of efficiency was 55 per cent for 0.028 in. grain size and at 20 per cent slip, while, with the same grain size, the loss of efficiency was about 36 per cent at 40 per cent slip.

VIBRATION, ROLLING AND STEERING

Quite a number of interesting and useful papers have been published on the subjects of vibration, rolling and steering. The first, to which reference will be made, is by Mr. J. Dieudonné on the vibration of ships.[17] The paper deals mainly with actual observations on the hull vibrations of ships, and since the accurate calculation of the frequencies of vibration of the hull in the early stages of the design is important, naval architects will appreciate the approximate formulae given for calculating these frequencies. Thus for two-node vertical vibration, the formula is

$$V=111.5\left[\frac{K^2 I}{\Delta L^3(1.2+B/3d)}\right]^{0.405}$$

In this formula, I is the moment of inertia of the midship section and Δ, L, B and d have the usual meanings. The factor K depends upon the distribution of the moment of inertia along the length of the ship. For ships without superstructures K = 1. Where there is a superstructure the K value has to be calculated according to a method described in the paper and it takes into account the contribution of the various decks and their extent. For transverse vibrations the formula for two-node vibration is

$$T_1=76K'\sqrt{\frac{I_t}{\Delta L^3(1+d/B)}}+29$$

in which I_t is the moment of inertia about the centerline and K′ is a coefficient calculated in the same way as K.

In the realm of ship motions a most instructive piece of theoretical research work has been carried out by Mr. L. J. Rydill, R.C.N.C.[18] This deals with the steering of ships in waves, and, in the first place, the well-known linear equations for the unsteered motion of ships in calm water are extended to apply to the unsteered motion in regular waves, using the Froude hypothesis to evaluate the excitation. Two methods of dealing with the steered motion of ships among waves are employed; the first uses differential equations and demonstrates the ineffectiveness of the rudder in reducing the angular deviation from course when the frequency of encounter with the waves is high; the second uses servo mechanics and demonstrates how a control system can be designed so as to reduce the high frequency movement of the rudder while retaining an adequate margin of directional stability. Commenting on the application of the theory, the author states that the linear theory does not demonstrate adequately the difficulty of course keeping in following seas. He suggests that the tendency to broach-to may be partly due to the action of the wind which is not allowed for, while, in addition, there may be terms of both linear and non-linear type omitted from the analysis which have a significant effect.

Experiments have been carried out in the United States of America to determine rudder forces and torques.[19] These measurements were made during the trials of the U.S.S. *Norfolk*. In the course of these trials, both rudder forces and rudder torques were determined. The procedure adopted was to measure the rudder torque and to measure the rudder forces on the upper and lower bearings. From these measurements it was possible not only to deduce the force on the immersed part of the rudder, but also to determine the vertical and horizontal centers of pressure. Records were taken at speeds of

25 and 30 knots, and a third set of results for a speed greater than 30 knots. Measurements were made every 5 deg. of helm up to a maximum of 35 deg. The greatest measured force was 563,000 lb. and the greatest moment was 601,000 lb. ft. The results were compared with model experiments and calculations, and it appears that model experiments are adequate for determining design lift and drag coefficients, but they appear to give torque coefficients which are lower than those observed on the full-scale trials. Joessel's formulae are adequate for predicting normal forces, but not for predicting torques.

Professor A. M. Robb has made a theoretical contribution to the problem of rolling of ships.[20] In this analysis he discards the generally accepted assumption that the righting lever can be written GZ = GM sin θ and substitutes the "wall sided"

$$GZ = GM \sin\theta \left(1 + \tfrac{1}{2}\frac{BM}{GM}\tan^2\theta\right).$$

This expression is reduced to

$$GZ = GM\left(\theta + \tfrac{1}{2}\frac{BM}{GM}\ \theta^3\right) \text{ or}$$
$$GZ = m(\theta + a\theta^3)$$

Using this value for the righting lever the differential equation of unresisted rolling in still water becomes

$$\frac{\Delta}{g}K^2\frac{d^2\theta}{dt^2} + \Delta m\,\theta + \Delta m\,a\,\theta^3 = 0.$$

This motion is no longer simple harmonic and hence the period of oscillation is not independent of the amplitude. This modified period T_L can be written in terms of the isochronous period T as follows

$$\frac{T^2}{T_L^2} = 1 + (\text{constant} \times a \times \theta^2),$$

where the value of the constant is never far from 0.73. Professor Robb has extended his investigation to the case of resisted rolling among waves where the resistance varies as the angular velocity. A comparison between the theory put forward and available model data on the subject shows reasonably good agreement.

An unusual aspect of the ship motion problem is provided by a ship at anchor, and in this connection we draw attention to a recent investigation on the riding qualities of light ships.[21] The investigation was undertaken because of experience with light vessels in the approaches of the River Hooghly, at the head of the Bay of Bengal. The factors which lead to parting of light vessels' cables and uncomfortable riding behavior were examined by carrying out a series of model experiments at the Ship Division of the National Physical Laboratory on behalf of the Commissioners for the Port of Calcutta. Two vessels of widely differing characteristics and weight distribution, but of approximately the same principal dimensions, were tested. The results of these tests have shown that differences in form and weight distribution have very marked effects on the cable tension experienced. Thus, in the light vessel *Fame*, at a current speed of 5 knots, the tension in the cable approaches the breaking load of short open-link iron cable. In an improved design (the light vessel *Flame*) the cable tension never exceeds about 75 per cent of this load, which, for Tayco cable, gives a factor of safety of about two in the worst test conditions. The introduction of a spring mechanism into the cable can reduce the tension by as much as 30 per cent. Another factor which reduces cable tension is damping of the motion of the bow, which can be produced by appendages such as deep bilge keels.

In this section we make reference, finally, to a subject not directly connected with the ship itself, but which is important from the point of view of those who travel in ships. This concerns noise reduction and a paper on this subject has been presented in the United States of America.[22] In this paper the author is concerned with the noise arising from ventilating installations, and a method of calculating the noise level is given which apparently agrees quite closely with observations actually made on board ship. Much of the noise in ventilating systems comes from the fan impeller on the motor armature. There is also aerodynamic noise arising because of unsatisfactory blade and housing design. A third type of noise is that caused by standing waves in the ducting. Standing waves will arise when any duct dimension is equal to a half wave length or some multiple of it. The wave length is the speed of sound in the duct divided by the frequency, which is usually the blade tip frequency. One other point which should be mentioned is that the vent system should be designed so that no outlet in a system serving quiet areas should also serve noisy areas.

DESIGN AND GENERAL TOPICS

The Amos Ayre Lecture for 1958 contains some useful data on the evolution of the cargo ship.[23] In this paper Mr. J. Ramsay Gebbie compares the features of the cargo ship in the earlier twenties with those of the present day. It is interesting to note that while the ship of 1920 had a speed of about 10½ knots, its counterpart today may have a speed of up to 14½ knots. The corresponding powers are 2,600 and 6,800 B.H.P., respectively, and although the power is more than double, the machinery weight has only increased from 754 to 900 tons. The factors which have brought about the change in ships over this period are developments in hull form and propellers, improved structural design, including welding, improved cargo-handling appliances and, not least, the vast changes which have come about in the accommodations.

Some design considerations relating to tankers have been put forward in a paper by M. Jean Perrachon.[24] These, though largely concerned with the

machinery of this class of vessel, are worthy of study by naval architects as the choice of machinery type has important effects on the general design of the ship.

Another special class of ship to which attention has been directed is the ocean-going ore-carrier.[25] In this paper the subject of design is considered from the economic point of view. The author presents data in such a form that the principal dimensions of a ship can be readily obtained, and also provides a method for finding the optimum power and speed. From the information given, it is possible, also, to determine weight and costs. It is also possible to investigate departures from conventional design.

Tugs have been the subject of a paper by Mr. E. C. B. Corlett, Mr. J. Venus and Mr. C. H. Gibson.[26] The tugs described were for use in the Manchester Ship Canal. Table II gives some idea of the principal features of these vessels.

TABLE II

Length B.P.	88 ft. 0 in.
Breadth molded	24 ft. 4 in.
Depth molded	11 ft. 6 in.
Load draft, aft	11 ft. 8 in.
Load draft, forward	7 ft. 6 in.
Total displacement, tons	294
Block coefficient	0.478
Steel weight, tons	133
Wood and outfit, tons	31
Machinery weight, tons	79

The paper is full of useful information and is worthy of study by designers of small craft.

We would draw attention to another special paper, this time on yacht design.[27] This deals with various materials for construction, including steel and iron, aluminum and plastics. It also discusses engines and auxiliaries and sails and rigging in sailing vessels.

The use of plastics in shipbuilding has become important in recent years, and the Southern Joint Branch of the Institution of Naval Architects and Institute of Marine Engineers held a useful symposium on the subject in the spring of 1958.[28] In all, there were six papers in this symposium, by different authors, and the subjects discussed were "Plastic Materials," by Mr. V. E. Yarsley; "Plastics in Ships' Hull and Accommodation Spaces," by Mr. J. West; "Plastic Boat Building," by Mr. G. Wood; "Plastics in Marine Engineering," by Mr. J. F. L. Ludgate; "Plastics in Marine Electrical and Electronic Systems," by Mr. J. M. Fleming, and "Decorative Laminated Plastics in Ships," by Mr. H. S. Newcomb.

Cargo-handling appliances, in the form of cranes, have been discussed in the United States of America.[29] This paper describes the experimental installation of cargo-handling cranes on the motor vessel *Thomas Nelson*, one of the four "Liberty" ships converted by the Maritime Administration in their program of upgrading the Reserve Fleet. In this experimental installation both electric and hydraulic cranes were fitted. All the cranes were of the level-luffing, full-revolving type, mounted on tracks running athwartships between the hatches. It is of interest to note that the installation of this crane system increased the lightweight of the ship by 76 tons, and at the same time raised the center of gravity by 0.5 ft.

A much neglected subject has been dealt with in a paper by Mr. Ivan A. Gavriloff,[30] *viz.*, masts and rigging. The author analyzes the forces on the mast and its rigging and proposes a method for design. It is considered that this method is very suitable for the design of masts and rigging in warships, and that it will facilitate economical and efficient design.

The modern warship has been discussed by Mr. P. Gisserot [31] and in particular the question of space, which has become of much more importance than was previously the case. This has led to relatively large superstructures being fitted which has, in turn, brought its own problems. The question of accommodations is also involved in the problem of space. The author believes that there is a greater need for technologists and the necessity for building ships which can, to a certain extent, be adapted to the evolution of arms and equipment.

The Andrew Laing Lecture for 1958, of the North-East Coast Institution of Engineers and Shipbuilders, was devoted to nuclear propulsion of commercial ships, and was given by Sir John Cockcroft.[32] In this lecture the author has considered not only the technical questions involved in such an application of nuclear power, but also the economic question which must always be to the fore in any commercial enterprise. It would appear from this analysis that, at the present time, the nuclear-propelled ship could not compete with the normal type of vessel unless there were a marked rise in fuel-oil prices. This does not mean, however, that the subject should not be pursued; advances have been so rapid in many fields, in recent years, that the time may yet come when nuclear-propelled ships are the order of the day.

We conclude by making reference to a further paper by Mr. C. V. Manley on the losses of ships.[33] This concerns the losses of small ships and, first of all, it is interesting and satisfying to note that over the period from 1903 to 1956 there had been a steady decline in the losses of ships. The percentage of the total was 1.14 in 1903, and 0.24 in 1956. Where, however, there is less room for satisfaction is in the fact that the smaller ships seem to suffer more than the larger ones. Thus 68 per cent of the ships wrecked, and 48.5 per cent of those lost by fire, were less than 300 ft. long. With regard to collisions, some 81 per cent of the ships lost were less than 300 ft. in length. The statistics upon which these figures were based were compiled by Lloyd's Register of Shipping, and it is by a scientific exami-

nation of these figures, combined with technological progress, that we can expect, in the future, the sea to become even safer than it is to-day.

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A pin-sized glass rod with a magnetic coating has been developed as a combination switching and storage element for digital computers. When wired with 10 sets of winding per inch, the rod provides a memory element of a new order of compactness, achieving a packing density of 1,000 information bits per cubic inch. Requiring 20 milliwatts of energy to store one bit, the rod shows switching speeds up to 4 thousandths of one microsecond. The characteristic hysteresis loop is nearly square. The element is capable of reliable operation at temperatures up to 300°F. By virtue of its ability to perform as a logic switch, its use is expected to permit reduction in the number of semi-conductors required in a computer. Its small size and ruggedness suggest application in missile and satellite circuitry.

—from ELECTRICAL ENGINEERING, March, 1959

An ultrasonic seam welder is a recent addition in a rapidly growing field of ultrasonic technology. Westinghouse engineers have developed a welder which can weld sheets of dissimilar metals continuously. Sheets of the metal to be welded are passed between two wheels which vibrate at the rate of 20,000 cycles per second. Pressure of the wheels presses the metal tightly together. The vibration frequency of the wheels acts to break up the oxide coating on the metals and permit inter-atomic contact. No electric current is used, although the weld is similar in appearance and properties to an electric seam weld. A rate of 15 inches per minute has been achieved with aluminum sheets 0.010 inches thick. Further development is aimed at increasing the speed of the welder and the thickness of metal which it can handle.

—from WESTINGHOUSE ENGINEER, March, 1959

Two of the largest tankers yet to be completed in this country are the SS SANSINENA and a sistership recently completed by the Newport News Shipbuilding and Drydock Company. The 810 foot ships each have a full load displacement of 77,000 tons. The cargo capacity is nearly a half million barrels of oil at a designed speed of 17.2 knots. The 25,000 shp is provided by a geared turbine operating at 825 psig and 850°F at the throttle. The ships are the first two of three on order by the Barracuda Tanker Corporation. The third will be delivered later this year.

—from MARINE ENGINEERING/LOG, March, 1959

H. S. PREISER AND B. H. TYTELL

CATHODIC PROTECTION OF AN ACTIVE NAVY TUGBOAT WITH ZINC ANODES—42 MONTHS' SERVICE TEST

THE AUTHORS

Herman S. Preiser is a Corrosion Engineer for the Research and Development Division of the Bureau of Ships, Navy Department in charge of cathodic protection. He has been employed in the Bureau since 1950. He is a graduate of the New York State Maritime Academy and Webb Institute of Naval Architecture. He is a licensed Professional Engineer in the District of Columbia, and the State of Virginia. Mr. Preiser has been a frequent contributor of articles to the JOURNAL. *In addition to his membership in the American Society of Naval Engineers, he is also an active member of the Society of Naval Architects and Marine Engineers, the Electro-Chemical Society and the National Association of Corrosion Engineers. Recently, he was granted a patent on an insoluble trailing anode.*

Boris H. Tytell is a research chemist in charge of the Research, Development and Consulting Section of the Chemical Laboratory at the Boston Naval Shipyard. He has been intimately concerned with marine corrosion problems since he entered Government service in 1930. He received his B.S. degree in Chemistry in 1921 from Brooklyn Polytechnical Institute and spent two addtional years pursuing graduate studies there. He is an active member of the Electro-Chemical Society, National Association of Corrosion Engineers and The Society of Plastics Engineers.

INTRODUCTION

In JULY 1954, thirty zinc anodes with cast-in straps (specification MIL-A-18001, Type ZHS), containing less than 0.0014 per cent iron, were installed on the bottom of the Navy tug YTB-542. These anodes were welded on the bottom of the tug at the 3 foot waterline, 15 anodes on each side, evenly spaced from frame 5 to the after perpendicular. The tug had been painted in April 1954 with the standard Navy hot plastic paint system, comprising a vinyl butyral-zinc chromate-phosphoric acid wash primer (Formula 117), a phenolic-cumorone-indene-zinc chromate, anti-corrosive (Formula 14N) and a hot plastic antifouling paint (Formula 15HPN).

The first polarization potentials, using a silver-silver chloride reference electrode and a potentiometer-voltmeter, were taken by Boston Naval Shipyard laboratory personnel on 21 July 1954, several days after the tug was undocked. The readings obtained were in the range of 950 and 1000 mv. Weekly readings were then taken with the tug at rest in the harbor and periodic readings with the tug underway at about 7 knots. These readings were tabulated for a period of 15 months, until the tug was drydocked for overhaul on 1 November 1955. The details of the results obtained on the first drydocking of the tug have been published in the technical literature.[1] As a sequel the present paper summarizes the effectiveness of the cathodic protection

installation after 42 months of service at which time the zinc anodes were renewed and the hull repainted.

During the first hull inspection on 1 November 1955, the underwater hull area of the tug was found to be in excellent condition with only about 20 per cent of zinc consumed from each anode, except one zinc anode which showed no signs of deterioration whatsoever. The tug was therefore returned to service with the same anodes left intact and without a repainting, when the overhaul was complete in December 1955.

Periodic potential readings were resumed until the tug was again drydocked in January 1958, an additional period of 26 months. The potentials, referred to as silver-silver chloride reference electrode, of the ship taken at rest over a 21-month period are plotted in Figure 1. As shown, the protective potentials averaged about 810 mv. Minor depolarization to 780 mv occurred while ship was underway at speeds between 7 and 10 knots similar to the behavior reported in the earlier paper. Figure 2 is a graph of potential data recorded prior to the first drydocking and is included for comparison purposes.

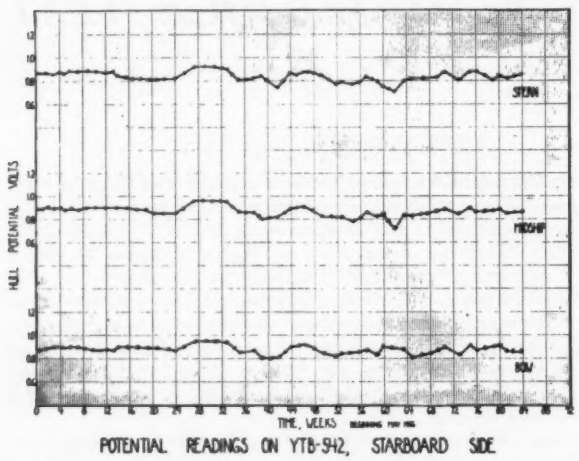

Figure 1. Graph of Hull Potential of YTB-542 (Ag-AgCl) Versus Time. Readings Were Taken on Starboard Side with Ship at Rest from May 1956 to January 1958.

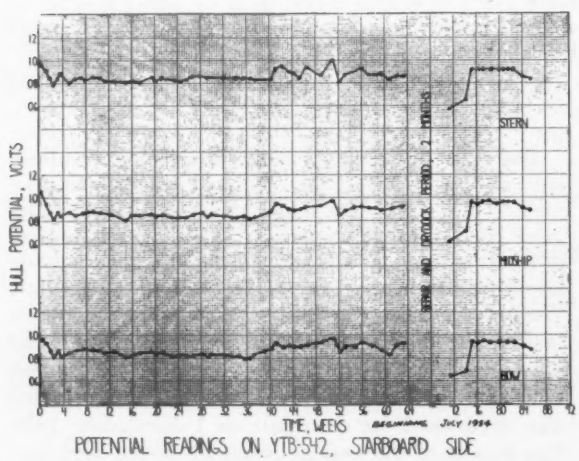

Figure 2. Graph of Hull Potential of YTB-542 (Ag-AgCl) Versus Time. Readings were taken on Starboard Side with Ship at Rest from July 1954 to December 1955.

OBSERVATIONS ON DRYDOCKING

An examination of the underwater hull immediately after the second drydocking disclosed the following:

(1) The bottom paint system was still in good condition, with the exception of a few small areas.

(2) About 75 per cent of the zinc on each anode was consumed during the 41 months of the tug's operation afloat.

(3) No corrosion was observed on any part of the tug's bottom, rudder or propeller.

(4) Soft, easily removable, corrosion products were found on the surfaces of the anodes.

(5) Two zinc anodes, one of which was originally reported, showed negligible corrosion and retained substantially their original dimensions.

For purposes of comparison, the condition of the hull and zincs, photographed at the same approximate location at progressive intervals extending over the test period, is shown in Figures 3, 4, and 5. The general condition of the bottom paint and the appearance of one inactive anode at the end of the test is shown in Figure 6. The appearance of the stern, rudder and propeller after 15 months and 42 months of service is shown in Figures 7 and 8.

DISCUSSION

The experiment with zinc anodes containing 0.0014 per cent iron or less, on the bottom of the tug during an operating period of 41 months establishes this material as an excellent galvanic anode for the cathodic protection of ships, active or reserve. It proves that zinc of such high purity con-

Figure 3. Arrangement of Zinc Anodes on Hull (Port Side Aft), July 1954.

tinues to provide current for cathodic protection in sea water (Boston Harbor water resistivity, 40 ohm-cm) as long as there is zinc metal left. While it was found in laboratory experiments that low iron, high purity zinc anodes form polarizing surface films with time, the current output was not affected substantially by the soft porous corrosion products found on the surfaces of the anodes after 42 months of service. While aluminum alloys as well as magnesium alloys may be considered suitable materials for galvanic anodes, a comparison of the properties of each of these galvanic materials may guide corrosion engineers designing cathodic protection systems in the choice of the proper material.

Figure 4. Appearance of Zinc Anodes and Hull after 15 Months' Service (December 1955)—Port Side Aft.

Figure 5. Appearance of Zinc Anodes and Hull after 42 Months' Service (January 1958)—Port Side Aft.

AMPERE HOUR CAPACITY

The ampere hour capacity per pound of zinc (specification MIL-A-18001) has been found to be 350 at an efficiency of about 95 per cent. Aluminum with a zinc content of 5 per cent (maximum trace impurities, per cent: iron 0.17, copper 0.02 and silicon 0.10), produces 725 amperes per pound at reported efficiency of 56 per cent. One pound of magnesium anode material (specification MIL-A-21414 (SHIPS)) has an ampere hour capacity of 500 at an efficiency of 50 per cent. This efficiency is true only when the current demand on the anodes is 100 per cent of their ability to produce. It can be seen, therefore, that materials with a low efficiency rating are going to lose more metal through self corrosion at low-current output, than materials with a high efficiency rating. Thus, zinc with an efficiency of about 95 per cent loses little weight through self corrosion regardless of the current demand.

Figure 6. Appearance of One Inactive Zinc after 42 Months' Service (January 1958).

CURRENT OUTPUT PER SQUARE FOOT OF ANODE

Based on a standard anode size of 12″ x 6″ x 1¼″, aluminum and zinc both have an initial current output capacity of one ampere per square foot. Magnesium with three square feet of exposed area, has a current output of 2.0 amperes per square foot. In time this current output drops to about 0.8 amperes per square foot of zinc, 0.5-0.6 amperes per square foot for aluminum, 5 per cent zinc. Magnesium current output remains substantially constant with time.

DRIVING POTENTIALS

The driving potentials of zinc and aluminum are 250 and 200 mv, respectively, against a polarized cathode of 800 mv. Magnesium has a driving potential of 700 mv against the same cathode. Because of the high current output of magnesium anodes through self corrosion and on demand, such anodes should be mounted on dielectric shields to prevent hull paint damage. The current output also can be limited by fixed resistors or controlled by a suitable rheostat. Zinc and aluminum anodes do not require such shields or other current controls.

Figure 7. Appearance of Zinc Anodes and Hull after 15 Months' Service (December 1955)—Stern View.

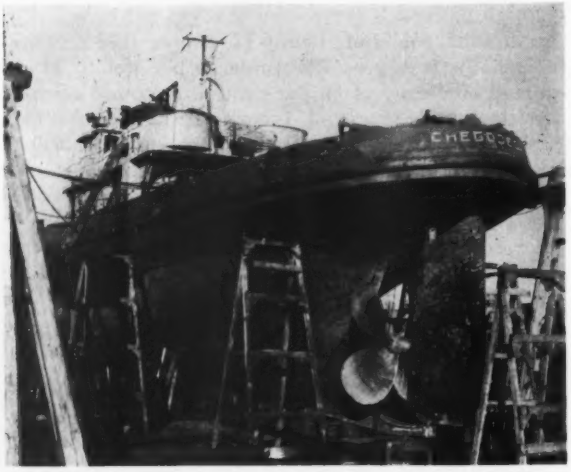

Figure 8. Appearance of Zinc Anodes and Hull after 42 Months' Service (January 1958)—Stern View.

WEIGHT AND BULK RELATIONSHIPS

The standard size zinc anode 12" x 6" x 1¼" weighs approximately 23 pounds, aluminum of the same size about 9 pounds and that of magnesium of the same dimensions would weigh about 5.4 pounds. Magnesium anodes of this size, however, are not generally used. Magnesium anodes 36" x 10" x 6½" are commonly used and these weigh about 135 pounds each (Type MHB, specification MIL-A-21414 (SHIPS)).

COST PER AMPERE-YEAR OF CURRENT

The cost of zinc per ampere-year based on an anode price of $0.20 per pound is calculated to be $5.20, that for aluminum, 5 per cent zinc alloy anodes at a quoted price of $0.60 per pound is $7.20 and that of magnesium at $0.40 per pound is $6.80. It must be pointed out, however, that this cost is true at 100 per cent current demand on the anodes. Self corrosion occurring during a period of low-current output will cause the cost to rise for aluminum and magnesium and very little for zinc.

CONCLUSIONS

From the observations on the performance of the zinc anodes with an iron content of less than 0.0014 per cent, on the bottom of the tug YTB-542, during a period of 41 months afloat (1 month consumed during drydocking), the following conclusions are drawn:

(a) Zinc anodes, specification MIL-A-18001, are suitable and desirable galvanic anodes for the cathodic protection of active and possibly inactive ships berthed at sea water sites.

(b) Such zinc anodes continue to provide protective current output despite the formation of soft porous surface corrosion products, as long as there is zinc metal remaining. The lack of current production from the two inactive zincs can be attributed to the existence of a poor bond at the strap embedment. This could be due to a casting flaw which sometimes occurs during a cold short and can be readily detected by means of penetrating dyes.

(c) The zinc anode cathodic protection system is self regulating and provides current on demand with a high order of efficiency.

(d) Cathodic protection systems using such high purity zinc anodes are economical.

(e) Reasonable protective potentials and adequate current capacities can be provided with zinc anode systems at very low bulk, so that turbulence in way of the anodes is minimized.

(f) Zinc with its 95 per cent efficiency is one of the most economical galvanic anode materials for active ships requiring moderate amounts of current for their protection.

(g) The cost of installation or renewal of zinc anodes is comparatively low for active or reserve ships of moderate size, for example, destroyer class.

ACKNOWLEDGMENT

The authors gratefully acknowledge the assistance of Messrs. F. E. Cook and R. C. Francis of the Bureau of Ships, and Messrs. B. Rozene and R. L. Burnett of the Boston Naval Shipyard in setting up and performing this experiment. The opinions expressed in this article are the personal views of the authors and do not necessarily reflect the official views of the U.S. Navy.

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"THE CLARKSON LETTER"

THE DISCONTENTED ENGINEER

ACKNOWLEDGEMENT

Hilbert Schenck, Jr., Assistant Professor of Mechanical Engineering, Clarkson College of Technology, is the author of this article. It was published in the January-February, 1959 edition of "The Clarkson Letter."

THE DISCONTENT of engineers in America is a fairly well-known phenomenon among managers and personnel experts. Surveys even suggest that engineering employees may constitute the most dissatisfied group in industry. The turnover rate of from 10 per cent to 20 per cent a year and the formation of "Engineering Unions," which now claim to bargain for 12 per cent of employed engineers, indicate the need for analyzing reasons for this discontent.

Such a state of affairs is variously greeted by puzzlement, derision or dismay by engineering lay people. Viewing the excellent starting salaries of engineering graduates, the ever-increasing number of engineers in management positions, and the wide public interest in technical matters, some citizens are tempted to shrug off technical man's distress as the whimpering of a prima donna. Others, thinking of days long past, still visualize the engineer as a single-minded misfit who can get along with nobody. Actually, surveys of the engineer show that he is perhaps the most solid of citizens. He votes conservatively, his divorce rate is almost nil, he is not deeply in debt, and his devotion to his home and the education of his children is exemplary by any standard.

Writing in the March 1957 issue of the *Journal of Engineering Education,* Professor H. A. Shepard notes that surveys suggest that the vast majority of working engineers want to become managers. Since only a relatively small proportion of any engineering group can achieve a responsible executive position, we might infer that discontent was the result of failure of the average engineer to achieve his goals. But such a simple explanation ignores certain other surveys and opinions. In *Electrical Engineering,* April, 1957, for example, a survey of graduating seniors in electrical engineering shows that 62 per cent of a 3400 man sample selected development engineering as their working goal following employment, while only 13 per cent chose manufacturing or production. The authors note "Here again the strong technical bias of students is shown. They have great interest in work that contributes to new things and ideas and therefore to the advancement of science and art of engineering."

I would like to suggest that many engineers, perhaps the majority of them, have as a goal, a combination of the two apparently divergent aims just noted. The engineer wishes to work on primarily technical and specialized projects. Almost all of his background and his formal training have equipped him for such a career. Most engineers are poorly trained to supervise large groups of people, particularly when compared with certain business or liberal arts graduates. The situation is somewhat analogous to selecting a president of a drug company. We might wish to appoint a medical doctor to this position, but we must realistically admit that few

doctors will have the proper training for such a job.

On the other hand, the engineer wishes to be treated as a management man. His difficult academic preparation, the insistent stress on his "professional" nature, and the wide publicity regarding his essential contributions to our well-being cannot help but instill in the engineer a feeling of self-importance, as well as realization of his unique capacities and talents. Since the majority of engineers function in an industrial situation, and since maximum status in this environment accrues to the manager, it is only natural that the engineer should aspire to management levels.

Unfortunately, the facts of economic and organizational life as now operative make it almost impossible for the engineer to reconcile this dual goal. As the engineer rises higher in the company hierarchy, he spends less and less time on the things that interest him most. A very few companies (General Electric is perhaps the best example) have attempted to create a double pathway to economic and status improvement. Those engineers equipped for personnel supervision, either by industrial training in college or by inclination, follow the traditional management ladder. Those not inclined in this way rise in salary and influence by purely technical excellence. How well this plan will work is difficult to say since it is a decidely new concept.

In most industries, however, the young engineer remains as a relatively undifferentiated employee. By consulting "job descriptions" as set forth by his company, the young man sees that above a certain (and often very modest) salary level, over half his time will be spent on employee supervision, job rating, purchasing and procurement duties, sales and marketing conferences, and similar non-technical duties. The engineer covets the added income and substantial prestige and influences connected with such positions. He dreads the difficult problems, the dreary conferences, and the endless form-filling duties for which he is unprepared, both mentally and emotionally. The result, I submit, is frustration and discontent. Further, the engineer who is essentially technically oriented and never achieves high status is unlikely to fully develop his special abilities. On the other hand, his promotion to managerial status simply means that he has been displaced from the crucial creative phase of technology to a position that would better have been filled by a trained business-school graduate. Thus the company with no special rewards for talented technical ability does not effectively utilize the engineering employee in either role.

One problem lies in the failure of the engineering societies to properly differentiate the engineer from his industrial partners. For example, the American Society of Mechanical Engineers has sections devoted to the purest of technical investigation and sections devoted entirely to the practice of management in the mechanical industries. As a result, the society cannot speak or act in behalf of any homogeneous group. When this problem is aired, some societies reply that the working engineer and the working manager share the same goals and ideals and thus can live beneath the same roof. This seems to be more wishful thinking than fact. The engineer who felt in tune with his non-engineering boss would not join a collective bargaining unit, yet more and more do so each year. Actually, the managing of a company whose products are engineering or are based on engineering is a specialized skill related to, but quite different from, the technical aspects of development and production. It would thus be entirely appropriate to break away from the major societies those sections devoted to management and form a separate group with this as its main object. The remaining members would then have a true technical society devoted wholly to the problems of the working engineer, and able to consider issues unique to this large group of skilled people. If a working engineer became a working manager, he would then simply change societies and thereby find a new group whose aims and views coincided more exactly with his own.

My belief is that the true engineering society should not engage in collective bargaining of any sort, for this is generally incompatible with creative endeavor. The true society should, on the other hand, carefully examine certain general problems and national issues as they relate to working engineers. The late lamented "engineering shortage" is an example of the lack of forthright leadership on the part of many of the societies. The president of one of the largest stated flatly (and with admirable accuracy) that a ten per cent decrease in gross national product would see "engineers coming out of our ears," but his was a voice in the wilderness. Most of the societies publicized the "needs" of the engineering profession and compared our own supply and Russia's. In view of the recent drop in engineering employment, one wonders if the employer members in the societies did not speak more loudly than the employee members. Discussions involving such issues as "engineers punching clock," the appropriate reward for discoveries by an employed engineer, and the addition of clerical help to relieve engineers of routine duties are difficult to carry on when both the person to be rewarded and the man who must pay for it are in the same organization. Yet it is in these very areas that the difficult and sensitive ideals of "professional attitude" and "professional development" are most often in doubt.

I believe that in the long run, the engineering executive who recognizes the dual requirements of status and the creative atmosphere in engineering jobs will not only obtain maximum utilization of his talent but will obtain it at little or no increase in cost. It is usually far cheaper to greatly reward the few individuals with honest talent, than to be continually forced into spiraling wage rises for every Tom, Dick and Harry with a degree and a sliderule.

ROBERT TAGGART

THE EARLY DEVELOPMENT OF THE SCREW PROPELLER

THE AUTHOR

is President of Robert Taggart Incorporated, a new firm organized to do contract research and development work in naval architecture, hydrodynamics, acoustics and propulsion. He received his Bachelor of Science degree in Naval Architecture and Marine Engineering from Webb Institute of Naval Architecture in 1942. He has served with the Consolidated Steel Corporation, U. S. Maritime Commission, U. S. Army Transportation Corps, the Navy Department and Reed Research, Inc.

EDITOR'S NOTE: *This is another in the series of articles by Mr. Taggart, in which he is exhaustively describing the development of marine propulsion.*

INTRODUCTION

TOWARD THE close of the eighteenth century there was some indication that the screw propeller was being seriously considered as a marine propulsion device. The submarine propellers devised by Bushnell and Fulton and the surface ship propellers proposed by Bramah, Lyttleton, and Shorter were early demonstrations that such a design was at least feasible. But the more popular schemes of Rumsey, Fitch, and Symmington far overshadowed the minor successes which were achieved with the screw propeller during this era.

The idea of a propulsion device resembling what is now called the screw propeller was certainly not new. The experience of the ancients with sculling oars coupled with the later development of rotary engines obviously suggested a combination of a series of inclined plates secured to a rotating hub. The aged windmill which derived rotary motion and torque from flowing air could work in reverse to impart motion to water with the input of a separately generated rotation and torque. The Archimedian screw pump had been in use for centuries and the fact that it could accelerate a mass of water was a well proven fact.

At first glance it seems somewhat curious that more of the early inventors did not resort to the screw for propulsion. The majority of devices which were coupled to the steam reciprocating engine, with the exception of the paddle-wheel, were far more complicated than most forms of screw propellers. The most probable explanation is that the screw was tried and found wanting. This could be due both to a lack of a proper screw configuration and the lack of an engine design which could take full advantage of the propulsive qualities of the screw.

At any rate it was not until the nineteenth century that screw propeller development got underway. At the beginning of that century it was considered a strictly second rate means of moving a ship through the water. By the middle of the nineteenth century it was rapidly replacing the paddle-wheel on the high seas. And by 1860 it was practically the only type of propulsive device to be installed in sea-going ships. The story of its rise to this eminent position is one of trial and error, of ridicule and contempt, of claim and counter-claim, and finally of the blending of diverse views into a workable and practical means for propelling ships through the water.

There appear to be three fundamental ideas of screw propeller design from which the modern propeller has evolved. These are the underwater windmill, the bladed wheel, and the Archimedian screw.

THE UNDERWATER WINDMILL

One of the earliest examples of the underwater windmill was proposed by Robert Hooke in 1683. Hooke actually used the device as an instrument for measuring the velocity of water currents but foresaw its potential for propulsion. It consisted simply of four radial arms extending from a central hub to which were attached flat, rectangular plates. The plates were inclined to the plane of the radial arms and propulsion achieved by rotation of the hub.

The basic principle employed was extremely simple; it was the same as the operation of a windmill in reverse. The characteristic action of the windmill, that of revolving when a wind force is applied in an axial direction, is based upon the reaction of a flat plate created by its deflection of the wind. As shown in Figure 1 the reaction force can be divided into two components, one parallel to the axis of the windmill, the other perpendicular to the axis in such a manner as to apply a torque to the shaft. The windmill-type propeller simply reverses this action, that is, a torque is applied to the shaft thus generating a force which accelerates the water.

As a propeller the revolving flat blade has several fundamental disadvantages. If the blades are rectangular they cannot be made wide enough to generate a reasonable amount of thrust unless their base is some distance from the hub; as this distance is increased the supporting radial arm must be made heavier to carry the thrust which is generated. The early windmill-type propellers had blades mounted at a fixed angle to the shaft axis; the inner part of the blade working at a low peripheral velocity was thus at too low a pitch angle and the outermost part of the blade working at a high peripheral velocity was at too high a pitch angle. This caused unequal distribution of thrust along the blade and caused parts of the blade to offer more drag than propulsion. Finally the flat plate working in the flow took no advantage of the hydrodynamic lift which could have been generated had the blade been given some sectional shape.

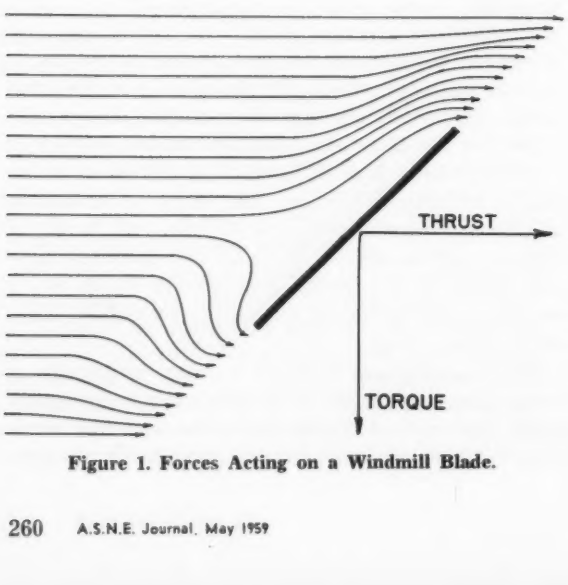

Figure 1. Forces Acting on a Windmill Blade.

The first engineer to experiment in a forthright and logical manner with screw propulsion was Colonel John C. Stevens of Hoboken, New Jersey. Colonel Stevens was fortunately endowed with adequate financial means, a keen engineering mind, and a great deal of patience.

In 1804 Stevens built a small boat twenty-five feet long in which he installed a rotary steam engine directly coupled to a screw propeller. His initial trials in the Hudson River showed the engine to be unsatisfactory. After replacing the engine with a type resembling that of James Watt he was able to make a cruising speed of four miles an hour with occasional spurts of speed up to eight miles an hour.

Stevens' propeller had four windmill-type blades and was built up of forgings to which were riveted iron plates as shown in Figure 2. Stevens increased blade width with the radius which permitted him to preserve the strength down to the hub. Also this varying width gave a more equitable distribution of thrust. The blades were adjustable and Stevens ran trials on the Hudson for about two weeks with a variety of pitch settings on the propeller. Apparently he considered the performance unsuccessful and redirected his efforts to paddle-wheel propulsion.

The difficulties which Stevens encountered were encountered by many of the early advocates of screw propulsion. The engines which he tried were of very low rotational speed. Turning at these speeds even the crude propeller which he had could attain only a small fraction of its available thrust. To Stevens this indicated that the paddle-wheel was a much more promising propulsive device for use with the prime movers which were available at that time.

It is regrettable that Stevens did not continue to devote his abilities to the development of higher speed engines rather than entering into what proved to be a fruitless competition with Fulton in the field of paddle-wheel propulsion. Had he pursued further his initial experiments and applied his talents to the development of a high speed prime mover the screw propeller might well have come into universal use at a much earlier date.

Many other inventors followed Stevens in the development of the windmill-type screw propeller. In 1805 in the "Annales des Arts et Manufactures d'O'Reilly" there is a memoir describing a ship with no less than eight windmill propellers mounted on shafts port and starboard which ran the length of the ship. John Millington of Hammersmith, England, patented a two bladed propeller in 1816. His blades were similar to those of Stevens and his method of mounting the propellers resembled that of Shorter, patented some years earlier. In 1824 M. Dollman of France added the idea of contra-rota-

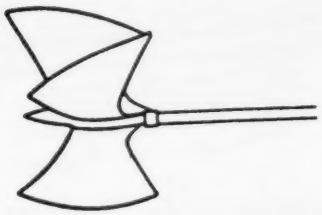

Figure 2. Colonel Stevens' Propeller and Original Blade.

tion. He used two-bladed windmill-type screws revolving in opposite directions on the same axis.

Probably most popular of the windmill-type propellers was the screw installed on the steamer *Liverpool* shown in Figure 3. This four-bladed pro-

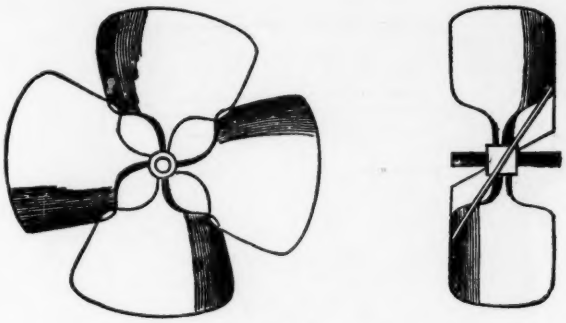

Figure 3. The *Liverpool* Screw.

peller was enlarged three times, each time improving the speed of the ship. It is evident that propeller design in this era was mainly a matter of trial and error.

A two-bladed screw of almost identical design to that on the *Liverpool* was patented by Joseph Taylor of London in 1838. And Captain George Smith of the Royal Navy patented contra-rotating two-bladed screws of similar blade shape in that same year. Smith's design is shown in Figure 4.

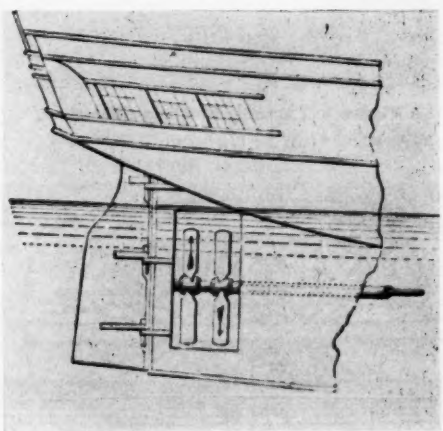

Figure 4. Captain Smith's Propeller.

At about this time inventors started varying blade shapes in attempts to gain better propulsive qualities from the windmill-type propeller. Peter Taylor, a rope merchant of Birching Bower, Lancashire obtained a patent in 1838 for the twin screw, four bladed design shown in Figure 5, in which the blade shapes were triangular. Another interesting feature of Taylor's design was the fact that the twin screws were rotated in such a manner that the blades intermeshed.

In 1840 Commander Edward John Carpenter of the Royal Navy patented the series of blade shapes shown in Figure 6. And David Napier of Millwall, England, proposed in 1841 the twin screw twelve-bladed design shown in Figure 7. The wheels were

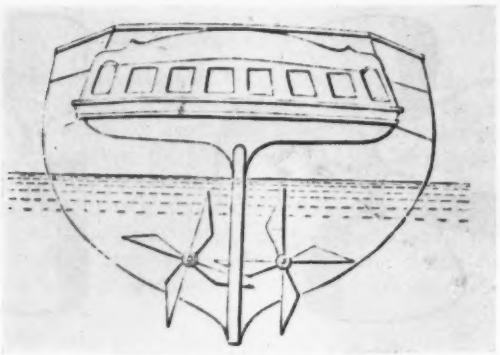

Figure 5. Peter Taylor's Propellers.

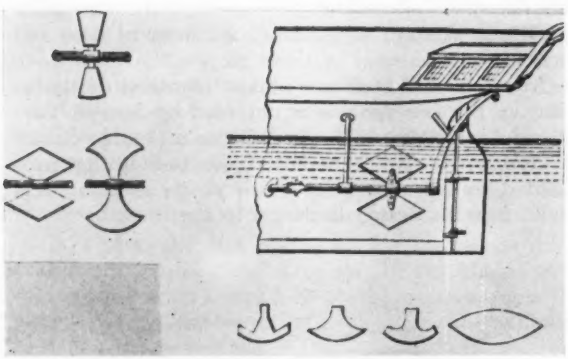

Figure 6. Carpenter's Blade Shapes.

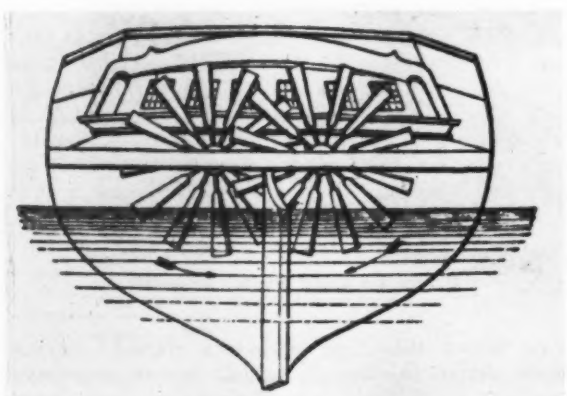

Figure 7. Napier's Propellers.

set with one slightly aft of the other and rotated in opposite directions. In 1843 Thomas Sunderland of London appeared to infringe on Carpenter's patent with the unique design shown in Figure 8.

These later variations of the blade shape and number of blades of the windmill-type propeller obviously showed little improvement over the basic type since they failed to remedy any of its fundamental faults. However, other inventors had been

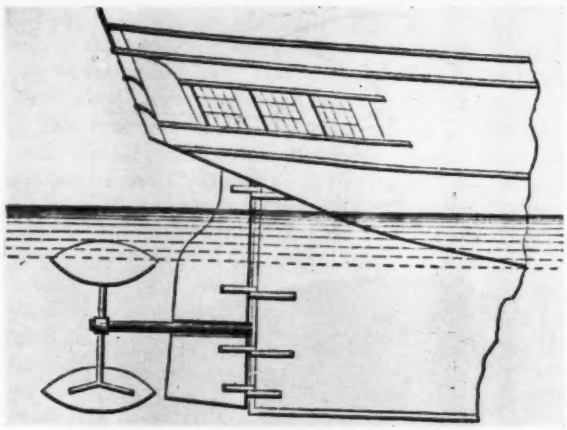

Figure 8. Sunderland's Propeller.

making progress in the development of the windmill-type into a workable propeller design.

In 1825 Jacob Perkins of London patented a contra-rotating windmill-type propeller. The unique feature of this invention, however, was the fact that Perkins applied a twist to the blades to give them a screw-like configuration. Each blade was set at an angle of 45° with the plane of the propeller at the hub flattening out to an angle of 22½° at the extremity. Perkins also patented a twin screw design with the same type of blades. In this latter design the shafts converged forward to be driven by a single engine shaft through gearing.

A similar idea was patented in 1830 by Clark Wilson of New Hampshire for driving a mill. And James Lowe of London in 1838 also referred to a twist in the blades of his propeller for the purpose of acting more like a screw to improve performance. His blade types were somewhat unique as shown in Figure 9.

George Blaxland of Greenwich, England, evolved another idea in 1840 for obtaining a varying pitch angle with increasing radius. His propeller blade, shown in Figure 10, was made up of three elements each set at a different pitch angle.

By 1842 the windmill-type propeller had developed into a really workable propulsion device. The wheel shown in Figure 11 was installed on the French mail boat *Napoleon* in that year. Several experiments were made with both three and four-bladed propellers. The one shown proved to be the best of the group. It had a pitch of 10.25 feet, a diameter of 7.5 feet, and a projected area ratio of 0.60. It is significant that during the experiments diameter, pitch, and projected area ratio were all varied as well as the number of blades.

THE BLADED WHEEL

The origin of the bladed wheel is somewhat in doubt. It originally resembled a wagon wheel with canted plates or blades substituted for spokes. This wheel design may have descended from some of the

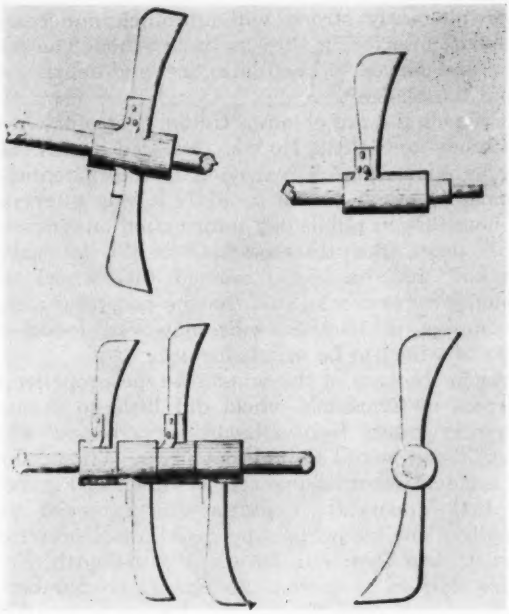

Figure 9. Lowe's Curved Blades.

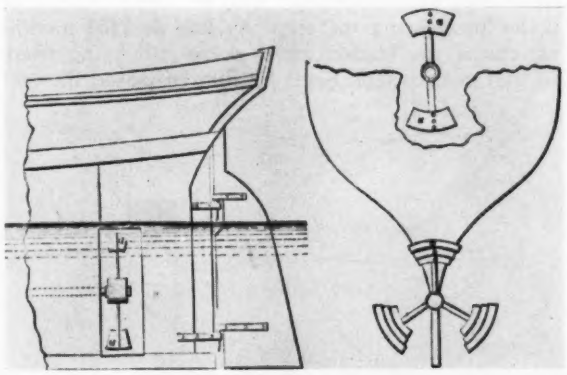

Figure 10. Blaxland's Propeller.

paddle-wheel types where the blades were canted 45° and the axis changed to fore-and-aft rather than athwartships. On the other hand it may have been a descendant of the Chinese or Moorish water-wheels which sometimes were built on an axial flow principal.

One of the earliest patents for a bladed wheel was issued to Henry James in England in 1811. His basic patent was for a stern paddle-wheel but he indicated in his claims that the blades could be set in obliquely and the axis run fore- and -aft to operate in the manner of sculling oars. A similar patent was issued to Benjamin M. Smith of Rochester, New York in 1829 although there appears to be no connection between these two inventors.

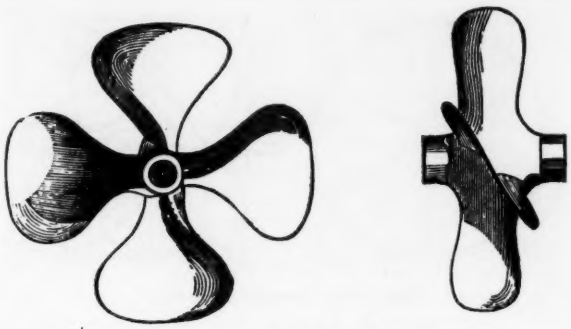

Figure 11. Propeller of the *Napoleon*.

Another patent was issued to John B. Emerson of New York in 1834 for a wheel in which the propelling blades or plates were supported by arms and encircled by rings. Emerson's original patent was made nearly valueless by the lack of clear specifications and drawings. However, he later filed new supporting sketches and was successful in a lawsuit for patent infringement against Hogg and Delameter who were utilizing a propelling device known as the "Ericsson Propeller."

These few examples appear to be the only pure illustrations of the original form of bladed wheel which like the windmill-type used flat rather than curved plates for blades. As in the case of the windmill-type the need for some form of helical shape in the blades was recognized at an early date. One of the first to see this requirement was Captain Delisle of the French Engineers.

In 1823 Delisle submitted a memoir to the Minister of Marine in which he proposed fitting out ships-of-the-line with four screws of five blades each. Delisle's propellers differed somewhat from the bladed wheel configuration of his day. The blades connected two wheels as shown in Figure 12. The blades were twisted in the form of a helix relative to the shaft axis. Two screws were to be placed at the bow and two at the stern. Plans were shown for unshipping the propellers so as not to impede the ship while under sail.

The French Government took no action on Delisle's memoir but later used it as a pretext for

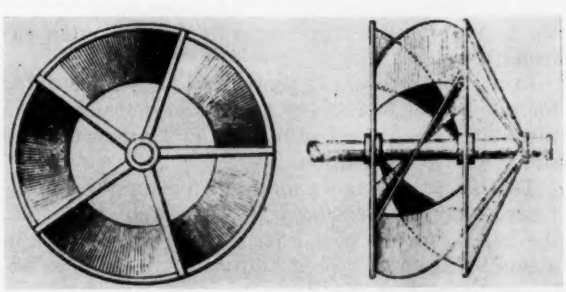

Figure 12. Delisle's Propeller.

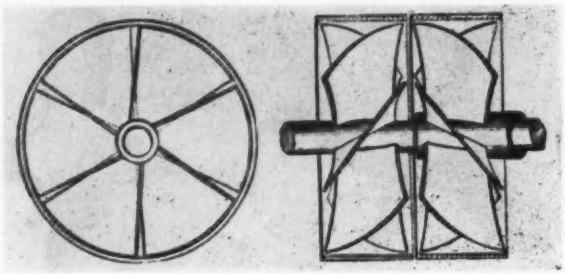

Figure 13. Church's Propeller.

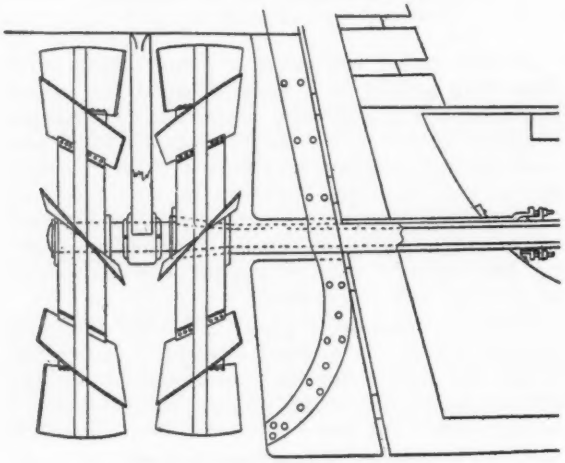

Figure 14. The Ericsson Screw.

evading Ericsson's patent in France where many wheels of this type were built.

William Church of Haywood House, England, took out a patent in 1829 for the contra-rotating bladed wheels shown in Figure 13. Church's blades extended from the outer housing or ring all the way to the hub and were formed in helical shape.

In 1836 John Ericsson, a Swedish engineer residing in London patented the contra-rotating bladed wheels shown in Figure 14. The blade ring of each wheel was supported from the hub by three twisted plates which formed an inner set of propelling blades. Around the periphery of the blade ring were mounted eight short, wide blades which also had a helical configuration and were tied together by a peripheral strap connecting their outer extremities.

In his contra-rotating propeller Ericsson provided for one wheel to revolve at a different speed than the other. His patent also covered twin screw and single screw installations.

This propeller shows many of the unique advantages of the bladed wheel. It was possible to obtain the increased thrust of a large number of blades in a small diameter without cluttering up the area adjacent to the hub. Yet both the inner and outer elements supplied propulsive thrust. The wheel design was inherently strong without much unnecessary material interfering with its basic action. The outer ring also served to keep lines, ice, and debris away from the blades.

Ericsson did not obtain a United States patent for his wheel until 1838. He was preceded in this country by Jesse Ong of Pennsylvania who patented an almost identical device in 1837. It was interesting to note that in publishing information on Ericsson's U. S. patent the editor of the *Franklin Journal* remarked that he had discussed this wheel with Colonel Stevens who said that he had tried a similar design in 1805. Stevens, however, found this type of wheel to be unsatisfactory.

As in the case of the windmill-type propeller the success of Ericsson's wheel did little to dissuade other inventors from attempting variations which they hoped would be improvements. William Joest of London patented the device shown in Figure 15 in 1841, apparently expecting the extended axial length of the blades to supply additional propulsive thrust. And Benjamin Biram of Wentworth, Yorkshire, felt in 1842 that increasing the number of blades to an even dozen as in Figure 16 would be helpful.

In 1845 Frederick Rosenborg of Hull, England, tried the modified blade shape shown in Figure 17. Rosenborg's arrangement also employed the propeller in steering the ship. As late as 1862 modifications of the bladed wheel were still being tried. In that year Joseph Scott Phillips proposed the odd

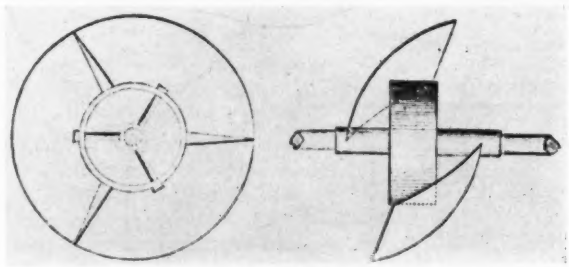

Figure 15. Joest's Propeller.

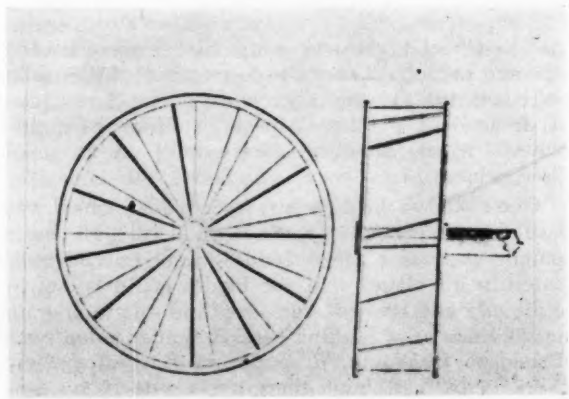

Figure 16. Biram's Propeller.

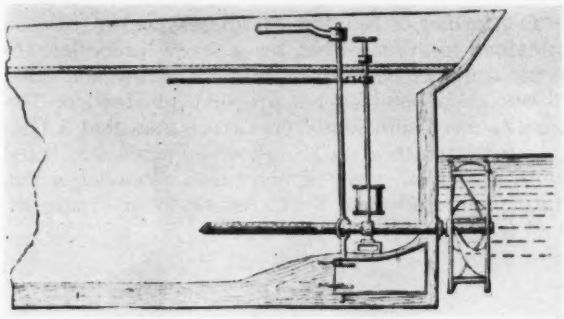

Figure 17. Rosenborg's Propeller.

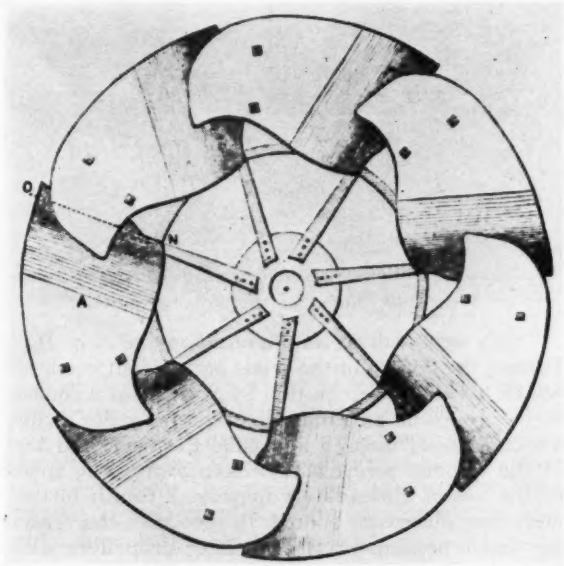

Figure 18. Phillips' Propeller.

configuration shown in Figure 18 which he called a "hoof screw propeller."

There is no clear cut evolution of the bladed wheel into the modern conception of a screw propeller although it possessed most of the elements of a successful propulsive device. It seems to have been used in the original Ericsson form and then dropped completely in favor of the conventional screw. It is probable that when improvements in metals and casting techniques permitted, the wheel built up of plates was simply abandoned.

THE ARCHIMEDIAN SCREW

The origin of the Archimedian screw and its capacity for imparting motion to fluids traces directly back to its inventor, Archimedes. Centuries of practical service as a pump gave a logical basis for its adaptation to propelling ships. It is most interesting that in the early methods of application the logical transition using an encased screw was not made. From the start the Archimedian screw was applied without a vital component—the casing. Without the casing the screw loses a great deal of its efficiency. However, due to the peculiar fact that almost any type of rotating blade will move a ship through the water the significance of the omission was lost.

One of the first nineteenth century inventors to suggest the Archimedian screw was M. Dallery of France. In 1803 he took a brevet of invention for an improved propeller which was a screw of two convolutions, one applied at the bow and the other at the stern. The screw swelled in diameter at the center and its length was twice its pitch.

In 1815 Richard Trevithick of Cambrone, Cornwall, patented a worm similar to the screw of Archimedes which was to have multiple threads wound around a fixed cylindrical hub. This screw could be applied at the bow, stern, or along the sides of the ship. When mounted at the bow it was to be made buoyant and move on a universal joint so as to accommodate itself to the unevenness of the waves.

In a memoir by M. Marestier published in France in 1824 several arrangements of Archimedian screws said to be used in America were shown. One of these consisted of a screw the length of the ship carried in a tunnel in the bottom. Also in 1824 M. Bourdon took out a brevet of invention for a propelling screw in which he recommended an expanding or increasing pitch.

Charles Cummerow of London patented an Archimedian screw in 1829. Figure 19 is a photograph of a model made from Cummerow's drawings. In the same year Josef Ressel, an Austrian, built an experimental steamship *La Civetta* at Trieste employing an almost identical screw. His eighteen meter boat was powered by a six horsepower engine and reputedly attained a speed of six knots. An explosion on *La Civetta* after a ten minute run injured several persons and ended Ressel's career in the field of marine propulsion. However, in German speaking countries he is generally credited with the invention of the screw propeller.

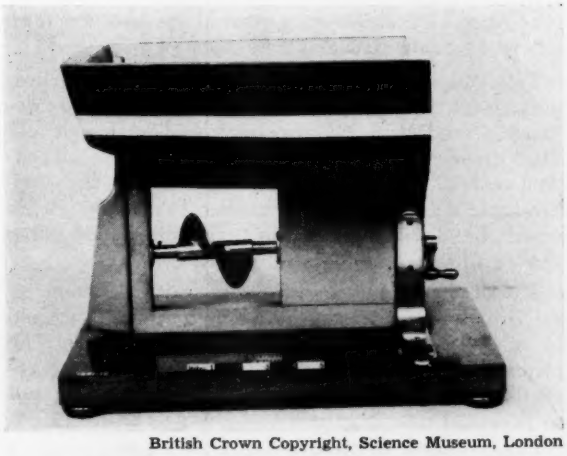

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Figure 19. Cummerow's Archimedian Screw.

Henry Ovinel of New York patented a pair of Archimedian screws for ship propulsion in 1830. His idea was to preserve the pump casing and to use screws of ship length both port and starboard. Also in 1830 Felix Peltier of New York in his propeller patent called for a screw with expanding pitch and with expanding diameter of the thread and John M. Patten of Milton, Pennsylvania, patented a "spiral or screw-wheel."

For the next twenty years numerous inventors obtained a series of patents on Archimedian screws. Had he been living that venerable Greek could have kept a large staff of lawyers busy with infringement suits. Some of these inventors were Bennet Woodcroft of Manchester in 1832, M. Sauvage of Paris in 1832, William Burk of Pennsylvania in 1834, Isaac Theal of New York in 1834, John F. Smith of New York in 1835, Edward Fitzpatrick of New York in 1835, Aretus Wilder of New York in 1836, Francis Petit Smith of Hendon in 1836, Frederick Fraisinet of Westminster in 1838, John Haddan of London in 1839, Miles Berry of London in 1840, and Commander George Beadon of the Royal Navy in 1845.

Most of these Archimedian screw inventors suggested little to improve the configuration of the screw for use as a propulsion device. Their main variations consisted of expanding the pitch of the screw from forward to aft, altering the diameter along the length of the screw, or employing additional threads. There were some curious minds, however, which examined more closely the motion of the screw and the surrounding fluid and saw means for improving performance.

One of the first of these was Robertson Buchanan, a Civil Engineer of Glasgow, who published "A Practical Treatise on Propelling Vessels by Steam" in 1816. Buchanan said, *"Experiments have been made on a kind of screw; but this, I believe, after a trial on a considerable scale in America was rejected. Some mechanics, however, still think favorably of it, and suppose that if a screw of only one revolution were used, it would be better than where a longer thread is employed."*

This statement by Buchanan is one of the earliest indications that the performance of the Archimedian screw could be improved by cutting it off. This theory was expanded upon by Thomas Tredgold in 1827 in his "Treatise on the Steam Engine." Tredgold conducted a mathematical investigation in which he demonstrated the *"impropriety of using screws of many convolutions."*

Josiah Copley of Pennsylvania applied some of the principles suggested by Buchanan and Tredgold in his invention of a water turbine in 1830. The propeller-type drive which Copley proposed resembled later screw propellers but was never put to practical use.

Francis Petit Smith discovered the advantages of a shortened Archimedian screw by accident. Smith was a farmer of Hendon, Middlesex, and in 1836 he obtained his first patent for a screw propeller. He built a model boat fitted with his screw and tried it out on a pond on his property at Hendon. The results were sufficiently encouraging so that a boat of six tons with a six horsepower engine was fitted with a wooden screw of two turns. A model of this boat, named *Francis Smith,* is shown in Figure 20.

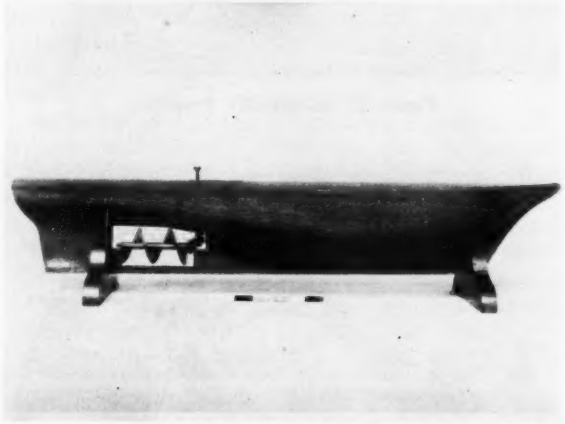

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Figure 20. Model of the *Francis Smith.*

Trials were run in the Paddington Canal in 1837. During the course of the trials an incident occurred which led to the recognition by Smith that a shorter screw would be advantageous. The propeller struck another vessel moored alongside the bank and half of the wooden screw was broken away. The speed of the vessel immediately increased. Smith fortunately was observant enough to recognize the reason for the improvement and his later propellers were limited to fewer turns of the screw thread. In 1839 he entered a disclaimer on his patent limiting his claim to the use of a single-threaded screw of one convolution or a double-threaded screw of half a convolution.

George Rennie, a civil engineer of London, also made a major contribution in the evolution of the Archimedian screw. Rennie combined the ideas of increasing pitch, multiple threads, and minimum convolutions in what he called a *conoidal* propeller patented in 1839. The elements of Rennie's helices were tangent to the surface of a cone which gave each of the three blades a pitch increasing from forward to aft. He shaped the blades to provide an easy entrance into the water giving them what would now be termed skewback. A model of Rennie's propeller is shown in Figure 21.

This propeller was installed on the 164 ton *Dwarf.* Driven by a 70 horsepower engine at 150 revolutions per minute the ship was said to have made a speed of 12 miles per hour. The propeller was 5 feet 10 inches in diameter and had a blade area of about 15 square feet. In 1843 this ship ran 200 miles in 23 hours with a calculated apparent slip of 12 per cent.

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Figure 21. Rennie's Conoidal Screw.

A similar design, but one of less apparent merit, was proposed by Charles Henry Joseph Forret of London in 1845. Forret's propeller, which he called *Devaine's Screw,* is shown in Figure 22. Another propeller of this type was that patented by John Samuel Templeton of Kensington, England, in 1846 which he called a *conoidal volute.*

Although the Archimedian screw in a wide variety of forms continued to be proposed for ship propulsion the final transition of this type of propulsion device to what is now recognized as a screw propeller was made by Rennie's conoidal screw. By the early 1840's the windmill-type propeller had evolved through a series of stages into identical forms of design. The bladed wheel began at this time fading out of the picture as a marine propulsion device.

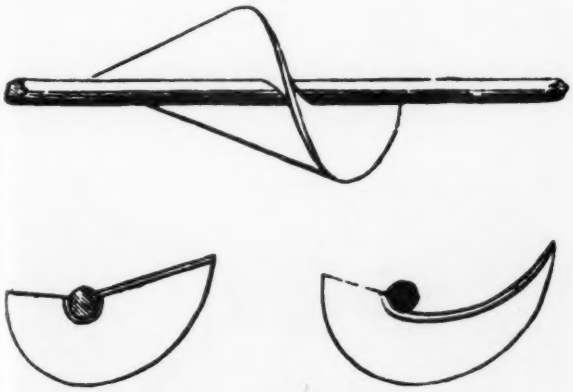


Figure 22. Devaine's Screw.

BYPATHS

While a definite progression of inventions was leading toward the modern concept of the screw propeller there were many deviations from the straight and narrow path which are worthy of note.

Commander George Beadon of the Royal Navy patented a series of Archimedian screw configurations in 1845 which are shown in Figure 23. Although none of these ideas were new at the time they do represent several ideas or means of employing this age-old device. The idea of using scroll arms rather than radial arms was based on the conclusion that the extremities of the screw were more effective than the portions closer to the hub. The addition of windmill blades at the extremities was intended to provide greater thrust.

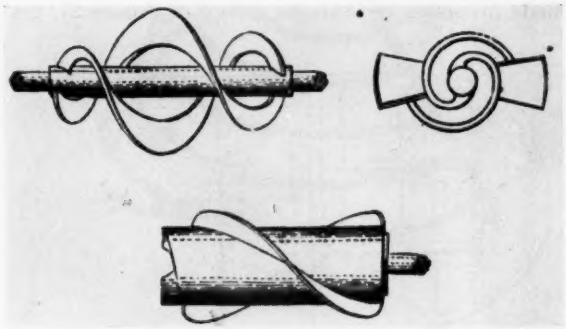


Figure 23. Beadon's Screw Propellers.

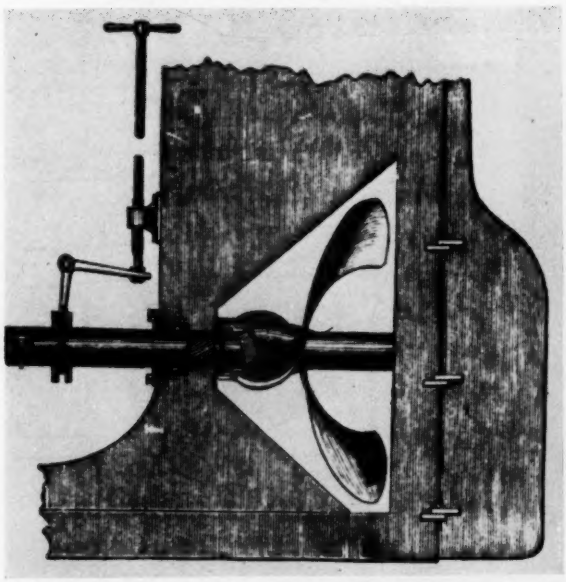


Figure 24. Beadon's Modified Design.

By 1851 Beadon had modified his ideas somewhat and in that year he registered the design shown in Figure 24. This was essentially the forward portion of a foreshortened scroll and was expected to be extremely effective.

The idea of using tandem interlocking twin screws was proposed by Peter Taylor of Hollingwood, England, in 1846. His design of two shafts with eight blades each is shown in Figure 25.

The problem of feathering the propeller when a ship was under sail was still prevalent in 1846. In that year John Buchanan of London patented the swiveling propeller shown in Figure 26. The blade was free to swivel through a small arc as limited by the slot in the shaft. When under sail the blade would feather automatically; when the shaft was rotated in either direction, ahead thrust would result. For backing, the shaft had to be turned to the position shown in the profile view. A rod was then inserted into the pivot shaft and a nut adjusted to lock the blade in one position.

John MacIntosh of London devised a flexible-blade propeller in 1847. As shown in Figure 27, the blades were built up of a series of thin plates in the fashion of a leaf spring. The blades were set at an angle of 45° to the shaft axis and when the axis was rotated they would tend to bend into the shape of a helix. It was anticipated that this propeller would adapt itself to a wide range of operating speeds the pitch being altered automatically by the applied torque.

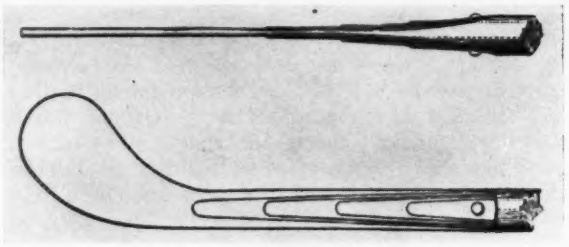


Figure 27. MacIntosh's Flexible Blade.

The similar but less elegant design shown in Figure 28 was proposed by Florid Heindryckx of Brussels in 1850. This latter inventor relied on the

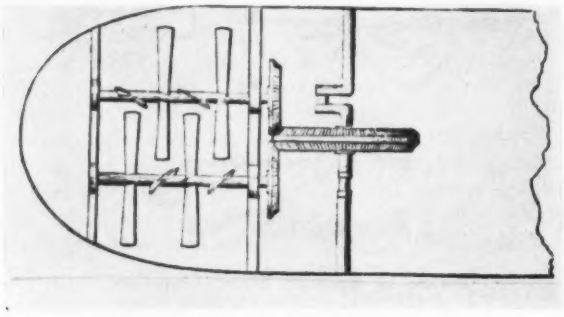


Figure 25. Taylor's Tandem Propellers.

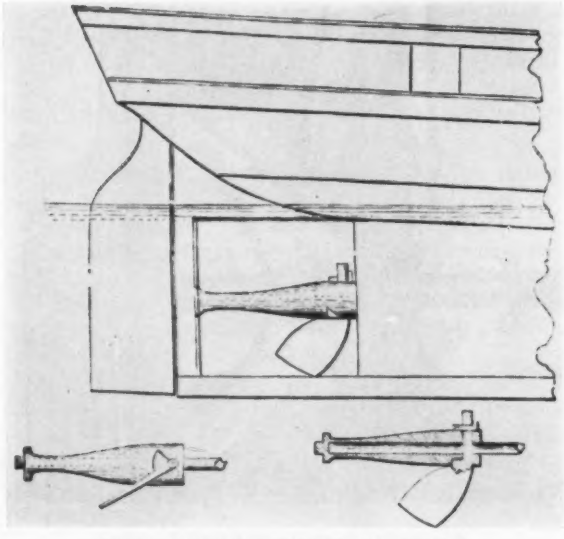


Figure 26. Buchanan's Swiveling Blade.

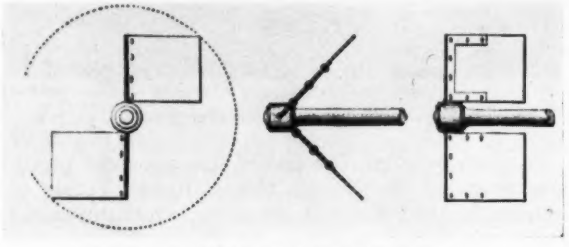


Figure 28. Heindryckx's Flexible Propeller.

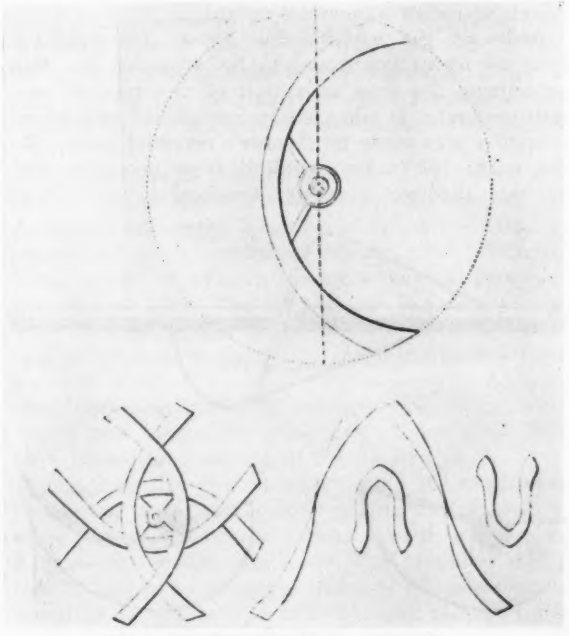


Figure 29. The Boomerang Propeller.

flexibility of single iron plates to achieve the flow adaptation necessary for optimum performance.

In 1848 Moses Poole patented the *Boomerang Propeller* shown in Figure 29. The idea for this design was communicated by Colonel Sir F. Livingstone Mitchell who was residing in Australia where he became intrigued with the weapons used by the aborigines. The wheel was so designed that the shaft axis was at the center of gravity of the blade. Any number of blades could be used with the supposed advantage that the working surface of each blade was removed from the hub area thus gaining higher efficiency. This propeller was applied to several large vessels of the day but performance reports were conflicting.

In 1853 Donald Beatson was concerned with the wasted energy in the water thrown away from the propeller in a radial direction by centrifugal force. To restrict this loss he designed the series of blade sections shown in Figure 30. A similar idea of a corrugated blade shown in Figure 31 was patented by Henry Walduck in 1854.

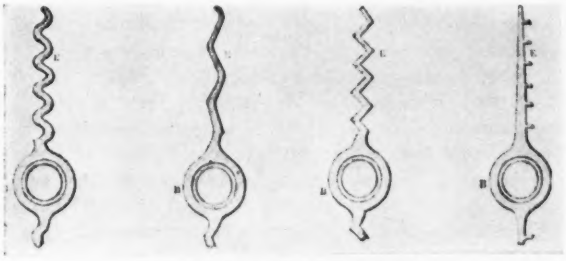


Figure 30. Beatson's Blade Shapes.

With the same idea in mind the Swedish engineer Charles Augustus Holm patented the propeller shown in Figure 32 in 1853. The leading edge of the blade was designed to enter the water without propelling and then the pitch was increased to infinity at the trailing edge. The periphery of the blade was flanged in a spherical curvature to limit the radial dispersion of water. A second flange was attached to the opposite face of the screw for aid in backing.

Holm's propeller design won a silver medal at the Paris Exhibition in 1855. It was installed on several ships but was concluded to be no more efficient than normal screw propellers due to the increased frictional resistance of the additional surfaces.

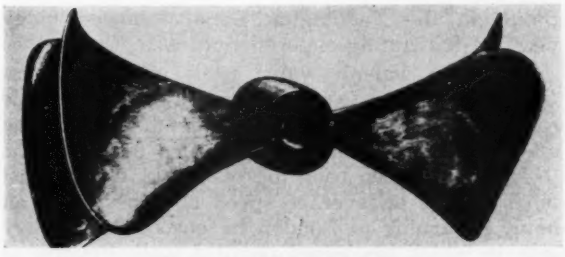


Figure 32. Holm's Propeller.

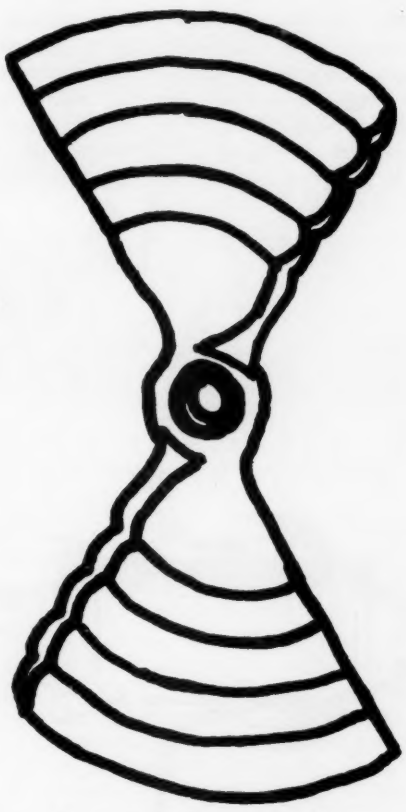


Figure 31. Walduck's Propeller.

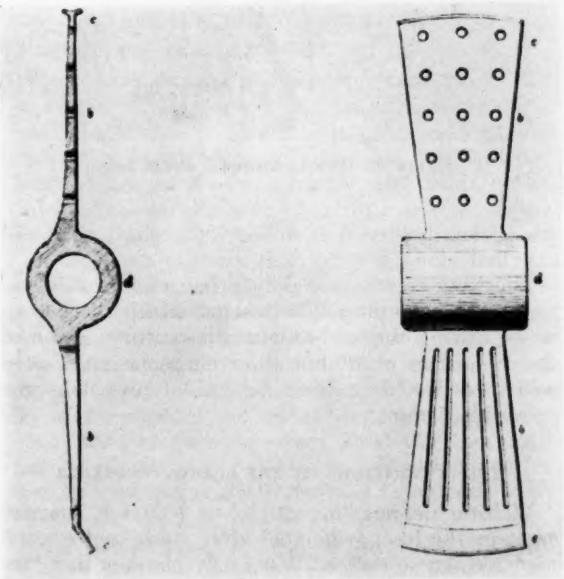


Figure 33. Fisher's Propeller.

The idea of a water retaining flange was also a part of the invention of John Fisher of Liverpool. In 1853 he patented the propeller shown in Figure

33. Fisher showed several configurations of his blades which were also perforated or slotted.

William Carwood, William Boaz, and Charles Colwell went even further in their ideas for limiting radial flow. Their 1862 patent was taken out on the design shown in Figure 34. Their design was intended to produce a *centripetal* action whereby the

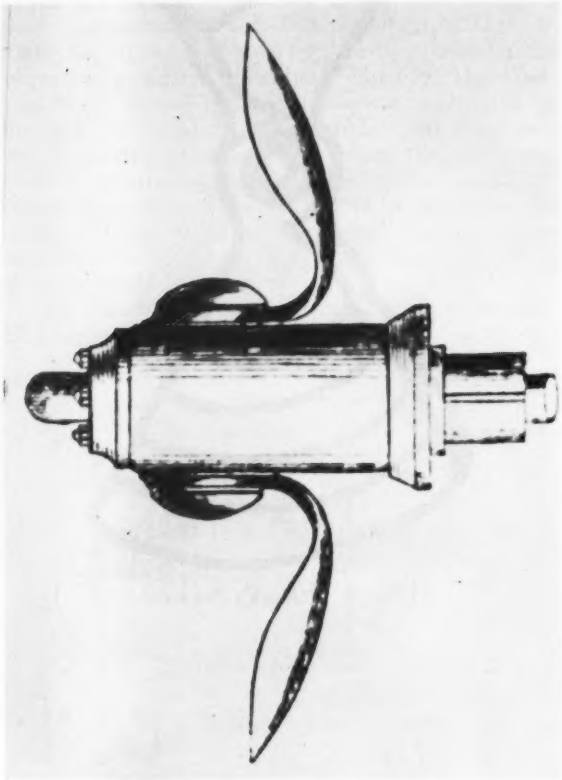


Figure 34. The Centripetal Propeller.

water was gathered in toward the shaft axis and expelled along it.

These, of course, are only a few of the wide variety of screw propeller designs which were proposed during the mid-nineteenth century. Some of the ideas had merit but their characteristics were seldom reflected in later designs of propellers for operational ships.

THE "INVENTION" OF THE SCREW PROPELLER

Webster defines "invention" as *a device, contrivance, or the like, originated after study and experiment.* When so defined it is fairly obvious that "inventions" of the screw propeller are legion. Therefore in the case of the screw propeller it seems necessary to employ a more restrictive definition if any major credit for its invention is to be given. This further definition might add that an inventor is one who after conceiving a device has the courage

FRANCIS PETIT SMITH

and tenacity to follow through to the point where his invention is commercially accepted.

When the more rigorous definition is applied the credit for the "invention" of the screw propeller narrows down to two men, Francis Petit Smith and Captain John Ericsson.

Francis Petit Smith was born at Hythe, in Kent, England, on 9 February 1808. Although he earned his living as a grazing farmer he was addicted to the construction of boats and propulsion schemes throughout his boyhood. The success of his model test at Hendon and of the trials of the *Francis Smith* in the Paddington Canal followed many years of frustrating experimentation.

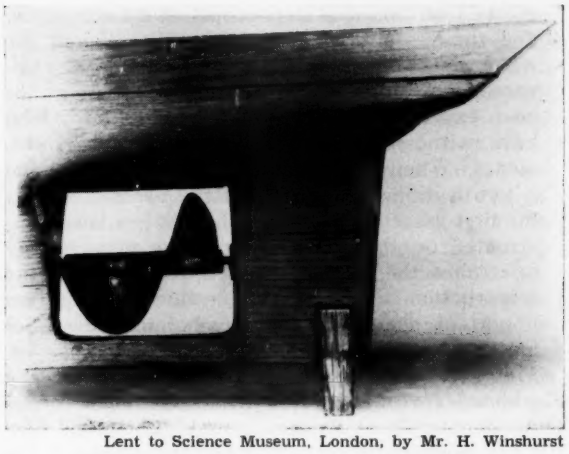
After the smooth water trials of the *Francis Smith* he decided to take the small boat to sea. He ran it from Blackwell to Gravesend to Ramsgate and to Dover. From Dover he sailed to Folkestone to Hythe and then back to Folkestone. On this last lap he made an average speed of seven miles an hour. Returning to London he encountered heavy seas in which his vessel handled well. Reports of this trip reached the Admiralty which demonstrat-

ed its interest by requesting a special trial of the vessel for inspection by the Lords of the Admiralty. The two trials conducted were favorably received and Smith was encouraged to build a larger ship.

Accordingly the *Archimedes,* a ship of 237 tons was built. She was fitted with a screw of one revolution as shown in Figure 35 and powered by two 45 horsepower engines. The ship was described in the *Inverness Courier,* a local newspaper, as follows:

"The *Archimedes* is rigged as a three-masted schooner, with her masts raking. Her length is 125 feet; average draught of water, 10 feet; capacity, 240 tons; power of engines, 80 horses.

"The mode of propulsion may be said to be by a portion only of the Archimedian screw. When the vessel was first tried, a full turn of that species of screw was employed. The inventor afterwards, for the sake of compactness, introduced the double-threaded screw, with half a turn of each thread, as more applicable to this vessel, although he prefers the other. This is of iron, and is fixed in an opening on the run of the vessel, above the keel, and about ten feet forward from the rudder The screw works transversely with the keel radiating the water all round as it turns with a backward movement. Its diameter is five feet nine inches, and the length fore and aft about five feet. It almost appears incredible that so small a portion of machinery could propel a vessel of such length; but the hold it takes of the water, and the velocity with which it turns, are the elements of its power. It is quite under the surface, and is therefore invisible to spectators, either on board or on shore. It is worked by a spindle forming its axle, which runs fore and aft and is connected with the steam engine, the velocity being acquired by a combination of spur-wheels and pinions. Each revolution of the larger wheel turned by the cranks of the engines gives, by the multiplied power, five and one-third revolutions of the screw, which consequently revolves at the rate of from one hundred and thirty to one hundred and fifty turns in a minute, according to the speed of the engine. In consequence of the powerful stream thus propelled against the rudder, the ship is actually found to obey the helm much more readily, and to be therefore more under command in steering, than either a common steam or sailing-vessel; so that she can easily turn around in one and a quarter or one and a half of her length, while it is well known that an ordinary steamer cannot do so with the paddles in less than six times her length. The shafts of the steam-engine work fore and aft, the cranks turning transversely, so as to communicate the power directly, by cog-wheels, to the screw; and there is one considerable advantage arising from this arrangement of the machinery,—namely, that the cylinders, and in fact the whole weight of the engine, rests immediately over the keel, where the vessel is the least liable to straining or twisting from the effects of undue pressure. The larger wheel is toothed or cogged with hornbeam (timber).



Lent to Science Museum, London, by Mr. H. Winshurst

Figure 35. The Screw of the *Archimedes*.

"The action of the screw is different from the operation of 'sculling,' in the particular that in sculling there are but two motions, the chief force being derived from the lateral; whereas the screw exerts an equal degree of power for every part of its surface towards the periphery in the direction of the radii. The successive columns of water, as fast as presented, are forced away from the mold-board of a plow. The action of the screw may be said to bear the same relation to 'sculling' which the use of paddle-wheels does to the ordinary mode of propulsion by oars.

"The '*Archimedes*' has made several trips and works well. Her speed is not quite so great as that of a first-rate steamboat in calm weather, but this is believed to result from the fact that her engines are on a new principle, and made by an inexperienced engineer. The full power of the boat is eighty horse-power, but in reality they do not work up to more than sixty.

"One of the greatest advantages of this invention, as applicable to all descriptions of shipping, is the circumstances that the screw may be thrown out of gear in two minutes and the vessel be put under sail alone. The screw is then turned by the motion of the vessel, but the drag is not more than half a mile in ten. Even the drag itself admits of being removed, as provision is made for totally unshipping the screw and bringing it upon deck.

"The advantages of the screw over paddle-wheels in ocean-steamers, it will be readily seen, must be very great. The leaning over of the ship often throws one of the paddle-wheels out of water and immerses the other too deeply. The screw is always in the water. The saving of fuel will be considerable, as the fires may be extin-

guished on board a ship propelled by the screw and the vessel used as a sailing-ship when the wind is full and fair. As a vessel of war the advantages would be palpable. This opinion has been expressed by officers of the royal navy who have witnessed the performance of the *'Archimedes.'* When it is recollected that this invention is yet in its infancy, and that the *'Archimedes'* is the first vessel on a large scale that has been constructed on the new principle, we may readily infer that the introduction of the screw in the construction of steamers is destined to work an important change in one of the most essential features of naval architecture."

The Admiralty looked upon the trials of the *Archimedes* with favor and arranged for her to circumnavigate Great Britain for demonstration. She did not, however, meet with universal approval. On one occasion a test of strength was entered into between the *Archimedes* and the paddle-wheel tug boat *William Gunston.* The vessels were tied stern to stern and the *Gunston* proceeded to tow the *Archimedes* astern despite the fact that the *Archimedes* was driving her screw ahead at full power.

Since the *Gunston* had engines of only 40 horsepower as compared with 50 horsepower for the *Archimedes,* this trial was looked upon as a demonstration of the superiority of the paddle-wheel. It was fairly evident that at that time the relative performance of paddle-wheels and screw propellers under high slip conditions was not understood. It was stated that Smith's invention was "theoretically most ingenious, but in practice deficient."

However, the adverse results of this one trial had little effect on the continued development and use of the screw propeller. In 1840 and 1841 the *Princess Royal,* the *Margaret,* the *Senator,* and the *Great Northern,* all screw-propelled merchant vessels, were constructed. The *Rattler,* originally intended as a paddle-wheel vessel was fitted with a screw to become the first vessel so propelled in the British Navy. A photograph of the propeller of the *Rattler* shown in Figure 36 illustrates the stage to which Smith's designs had advanced by 1843.

The Admiralty employed Smith's services in the design of a fleet of twenty screw-propelled ships. At the termination of the Russian war a screw fleet review was held at Portsmouth. His tour of government service completed Smith was made curator of the Museum of Patents at Kensington.

After the Portsmouth review the *Illustrated London News* of 26 April 1856 stated:

"The magnificent spectacle presented by our screw fleet at the naval review just concluded at Portsmouth, naturally directs public attention to the history of the screw propeller—an instrument which, though proposed centuries ago for the propulsion of vessels, has only lately been reduced to successful practice. Although many persons have aided in the introduction of this important improvement, it is mainly to the talent and persistency of one man that we are indebted for its practical realization; and rarely has it fallen to the lot of any one man, within his own lifetime, to confer upon his country benefits more timely or more momentous. That man is Mr. Francis Petit Smith. . . ."

CAPTAIN JOHN ERICSSON

John Ericsson was born in the province of Vermland in Sweden on 31 July 1803. A talented child with a mechanical bent he was producing his own designs and models at the age of ten. These attracted the attention of Count Platen by whose influence Ericsson was made a cadet in the corps of mechanical engineers and served as a leveler and draftsman during the construction of the Canal of Gotha.

In 1826 Ericsson constructed an air engine which he took to England to develop. By the time he turned his attention to the screw propeller he had the invention and construction of a steam boiler, a railway locomotive, steam fire engines, and a caloric engine to his credit.

Ericsson obtained his first propeller patent in England in 1836 and he made a model boat approximately three feet long on which to test the device. The model tests in a circular London bath were so

successful that he built a forty-five foot boat equipped with a pair of contra-rotating propellers; the name of the boat was the *Francis B. Ogden.*

An interesting description of this boat and its propulsion system was published in the *London Mechanics Magazine* in June 1837:

"The propelling apparatus is placed at the stern, and works entirely under water. It consists of a peculiar application of the old and well-known principle of the water screw, by which a great propelling power is concentrated in a small space. Of the degree of power concentrated no better proof can be adduced than the fact that the speed of 4½ knots, against wind and tide, was produced by an apparatus measuring only 5 feet 2 inches in diameter and 2 feet 2 inches wide, weighing only 615 pounds, and worked by a high-pressure engine having 2 cylinders of 14 inches stroke and 12 inches diameter, and which, during the experiment, made only 60 strokes per minute, and showed a pressure of not more than 50 pounds to the square inch. The new propelling apparatus consists of two short cylinders of thin wrought iron supported by arms of a peculiar form, which are placed entirely under the water at the stern and made to revolve in contrary directions round a common centre. To the outer periphery of each cylinder is attached a series of spiral planes or plates, which may be placed at any angle, according to the effect sought to be obtained, whether it be great speed or great propelling power.

"The apparatus may be made to ship and unship at pleasure; the engine that works it may also be loco-movable, so as to be worked upon deck and any part of the deck; and in these two peculiarities we are inclined to think the chief advantage of this new step in steam navigation will be found to consist. Sailing-vessels may by this means command all the aid that steam can give them without divesting themselves of any of their peculiar fitness for long sea voyages or undergo any change in their original construction."

Free running, the *Ogden* was said to have attained a speed of ten miles an hour. Ericsson saw in his propellers tremendous military potential. He approached the British Admiralty with the proposition of building larger, screw-propelled vessels but received little consideration. Undaunted, Ericsson arranged to take several representatives of the Admiralty aboard the *Ogden* for a demonstration. The discouraging results of this trip were given in a lecture by John O. Sargent before the Boston Lyceum in December 1843:

"The barge contained Sir Charles Adam, Senior Lord of the Admiralty; Sir William Symonds, Surveyor of the British Navy; Sir Edward Parry, the Commander of the Second British North Pole Expedition; Captain Beaufort, the Hydrographer of the Royal Navy; and other scientific and naval officers.

"In anticipation of a severe scrutiny from so distinguished a personage as the chief constructor of the British navy, the inventor had carefully prepared plans of his mode of propulsion, which were spread on the damask cloth of the magnificent barge. To his utter astonishment, as we may well imagine, this scientific gentleman did not appear to take the slightest interest in his explanations. On the contrary, with those expressive shrugs of the shoulder and shakes of the head which convey so much without absolutely committing the actor,—with an occasional sly, mysterious, undertone remark to his colleagues, —he indicated plainly that though his humanity would not permit him to give a worthy man cause for unhappiness, yet 'he could an' if he would' demonstrate by a single word the utter futility of the invention.

"Meanwhile the little steamer proceeded at a steady progress of ten miles an hour through the arches of the Southwark and London bridges towards Limehouse, and the steam-engine manufactory of the Messrs. Seward. Their lordships having landed and inspected the huge piles of the

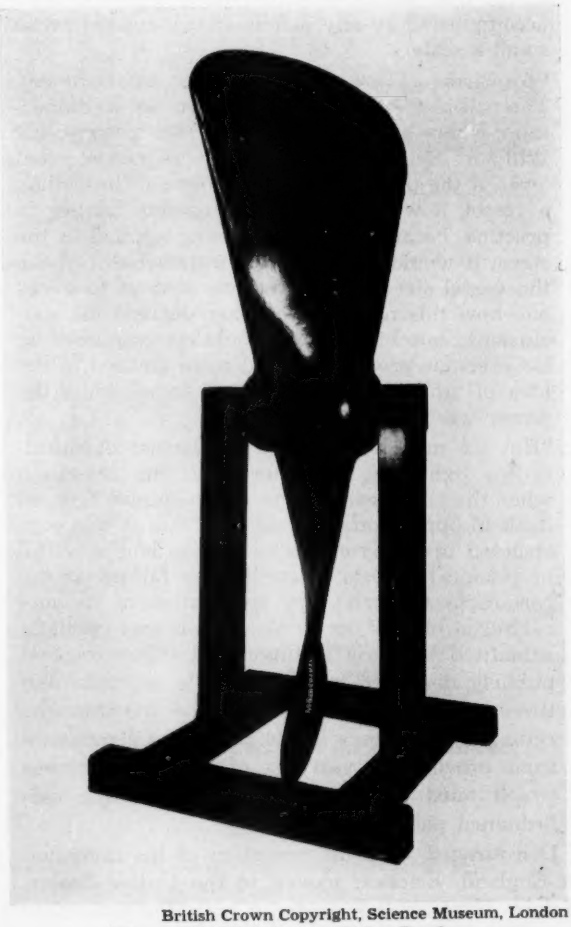


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Figure 36. The Screw of the *Rattler*.

marine engines intended for his Majesty's steamers, with a look at their favorite propelling apparatus, the 'Morgan paddle-wheel,' re-embarked, and were safely returned to Somerset House by the noiseless and unseen propeller of the new steamer.

"On parting, Sir Charles Adam, with a sympathizing air, shook Ericsson cordially by the hand, and thanked him for the trouble he had been at in showing him and his friends this interesting experiment, adding that he feared he had put himself to too great an expense and trouble. Notwithstanding this ominous ending of the day's excursion, Ericsson felt confident that their lordships would not fail to perceive the importance of the invention. To his surprise, however, a few days afterwards a letter written by Captain Beaufort, at the suggestion, probably, of the Lords of the Admiralty, was put into his hands, in which that gentleman, who had witnessed the experiment, expressed his regret that their lordships had been very much disappointed at its results. The reason was altogether inexplicable to the inventor; for the speed attained at the trial far exceeded anything that had been accomplished by any paddle-wheel steamer on so small a scale.

"An accident soon relieved his astonishment. The subject having been started at a dinner-table where a friend of Ericsson was present, Sir William Symonds ingeniously remarked that 'even if the propeller had the power of propelling a vessel, it would be found altogether useless in practice, because the power being applied in the stern, it would be absolutely impossible to make the vessel steer.' It may not be obvious to every one how this naval philosopher derived his conclusions; but his hearers doubtless acquiesed in his oracular proposition, and were amused at the idea of 'undertaking to steer a vessel when the power was applied in her stern.'

"But we may well excuse the British Admiralty for exhibiting no interest in the invention when the engineering corps of the empire arrayed itself in opposition to it, alleging that it was constructed upon erroneous principles and was full of practical defects; regarding its failure as too certain to authorize any speculation of its success. The plan of screw propulsion was specially submitted to many distinguished engineers, and publicly discussed in the scientific journals; and there was scarcely any one but the inventor who refused to acquiesce in the numerous demonstrations proving the vast loss of mechanical power which must attend the substitute for the old-fashioned paddle-wheel."

Discouraged with the reception of his invention in England, Ericsson moved to the United States. Before doing so, however, he built the passenger boat *Enterprise* for John Thomas Woodhouse and the *Robert F. Stockton* for Captain Robert F. Stockton, USN then on duty in London. The *Stockton* was taken to the United States under sail in 1839 and in 1840 she was sold to the Delaware and Raritan Canal Company and renamed the *New Jersey*. She saw service for many years as a steam tug on the Delaware and the Schuylkill Rivers.

The first screw-propelled vessel in the United States Navy was the *Princeton*, built in 1843, and it was claimed she was the first screw-propelled war-vessel ever built. The *Princeton* was designed and constructed by Captain John Ericsson. She was 164 feet long, had a beam of 30.5 feet, a depth of 21 feet, a mean draft of 17 feet, and displaced 673 tons. In October 1843, the *Princeton* was tried against the paddle-wheeler *Great Western* at New York and beat her.

Due in a large part to Ericsson's efforts the United States had 41 screw-propelled merchant ships operating by 1843. He also constructed the 2000 ton "caloric" ship *Ericsson* which was driven by a heated compressed air engine of four cylinders. These cylinders were 168 inches in diameter.

Probably the greatest service which John Ericsson rendered to his adopted country was the construction of the *Monitor*. Under the sponsorship of Donald McKay, Charles Ellet, and C. S. Bushnell, Ericsson built a model of the ironclad which was submitted to President Lincoln who immediately sanctioned construction of the full sized ship. At a cost of $275,000 the *Monitor* was built and delivered in 100 days with the resulting disastrous effect on the naval ambitions of the Confederacy.

In 1867 John Bourne said of Ericsson:

"Among contemporaneous engineers he certainly stands in the very first place, and to posterity he will leave a name that will never die. The possession of the services of such a man may form a fair ground of national congratulation; and America can never forget that at a critical period of her history he, like another Archimedes, stepped in, and by his ingenious devices, saved her navy from annihilation. The combination of that faculty of the imagination which we call invention, with the experience, the science, and the caution, which are the main qualifications of successful engineers, is a combination at once rare and precious; and like the talents of a great general it may become a power in the state that will influence its future destinies. Henceforth wars will be determined, not so much by a preponderance of muscle, as by a preponderance of brain; and the example of Ericsson shows how much may be done by one man to overturn existing systems of naval warfare, and to compel all nations to introduce others of greater efficacy."

CONTINUING DEVELOPMENT

Despite the successes of Smith and Ericsson there were still many problems to be solved in the design,

construction, and operation of screw-propelled ships. The early wooden-hulled ships were subject to heavy vibration and iron hulls were needed to resist the vibratory forces. With shaft and machinery below the waterline, stuffing boxes had to be developed to prevent leakage without damaging the rotating shaft. Thrust bearings were required to transmit the forward force exerted by the propeller to the hull. Higher speed engines had to be developed in order to realize the inherent efficiency of the screw and techniques for casting and machining strong, tough metals were needed.

The development of screw-propelled vessels in the United States by Ericsson had the advantage of more advanced marine engineering techniques than those developed by Smith in England. From the outset Ericsson used direct-drive high speed engines whereas Smith employed the slow speed engines used for paddle steamers and coupled them to the propeller shaft through speed increasing gears. Ericsson's engines still did not exploit the screw-propeller to the maximum extent possible, however.

As these many problems were gradually overcome and as higher and higher speed engines were developed, more and more screw-propellers were installed to supplement or replace paddle-wheels. The last of the great seagoing side-wheelers, the *Great Eastern* was fitted with a 24 foot diameter 36 ton screw-propeller which is shown in Figure 37. The 1600 horsepower screw engine of this mammoth ship shown in Figure 38 turned the propeller at 50 revolutions per minute. Both propeller and engine are a good example of the stage of development reached by this method of ship propulsion by 1858.

Although screw-propeller development had to await the accomplishment of many other techniques of marine engineering, it probably was delayed mainly by its own fundamental suitability as a marine propulsion device. As Sir William White once said: *"It take a first-class naval architect to design a really inefficient screw-propeller."* The truth of this statement is emphasized by the fact that the early propellers of Smith and Ericsson propelled ships in spite of their defiance of the more advanced principles of hydrodynamic design. Because even these inefficient models were competitive with the paddle-wheel the incentive to seek improvements in efficiency was not great.

By 1860 the screw propellers installed on most ships outwardly resembled the propeller of today. In this year it had attained undisputed mastery of the seas and the paddle-wheel gradually became obsolescent in seagoing ships.

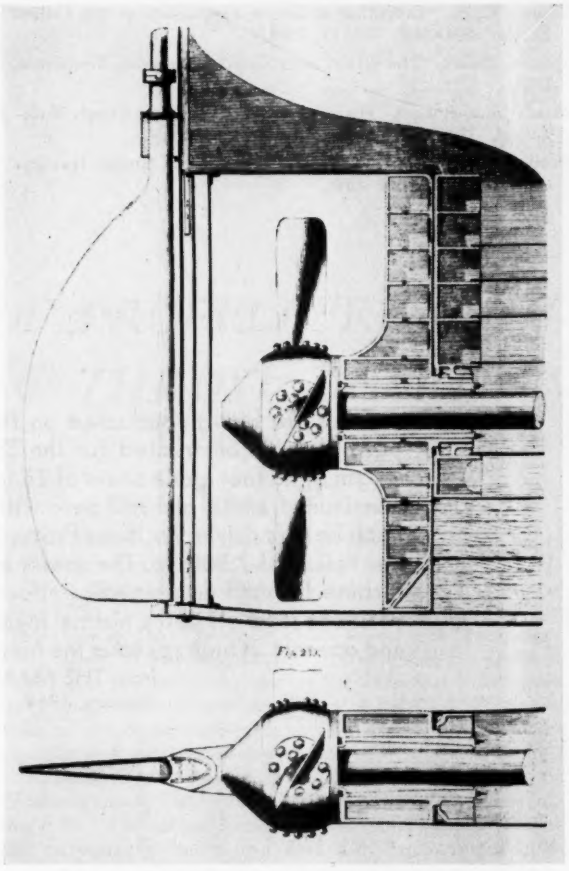

Figure 37. Original Screw of the *Great Eastern*.

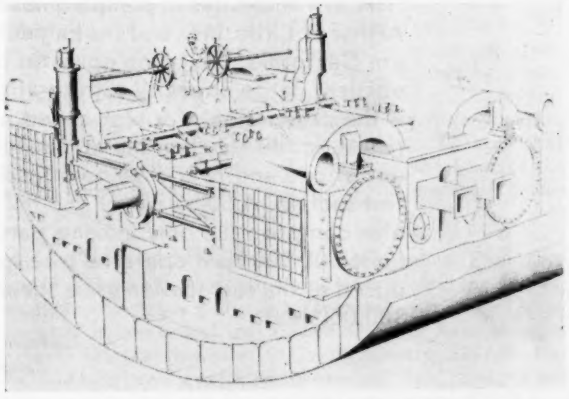

Figure 38. Screw Engine of the *Great Eastern*.

ACKNOWLEDGMENTS

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Trials are being conducted on the first of two new design high speed, patrol boats constructed for the British Navy. The 100-ton boats have a length of 96 feet and a beam of 25 feet. The hulls are constructed of welded aluminum frames, planked over with mahogany sheathed with fiber glass. Each boat is driven by three Proteus marine gas turbines having a continuous rating of 2,800 hp. The shafts are driven from the forward end of the turbine through an epicyclic reduction gear, at which point the shaft reverses to lead aft in the normal manner. The propellers are relatively small, and operate at high speed in the fully cavitating mode.

—from THE MARINE ENGINEER AND NAVAL ARCHITECT
January, 1959

A new technique in pumping has been developed jointly by personnel at Arthur D. Little, Inc., and the Engineering Center at the University of Southern California. The pump operates by rapid liquefaction of gas by a highly efficient, large capacity refrigerating arrangement. A striking application is the production of a high speed nitrogen stream for a hyper-sonic wind tunnel. In this scheme, the apparatus, called a cryopump, freezes nitrogen gas at one end of a chamber, causing flow from the other end through a test section. A 50 hp motor and the refrigerator compressor which it operates comprise the only moving parts, and provide results which would require 500,000 hp if operated by a gaseous diffusion pump. Conditions produced at the test section are a speed of Mach 20 at a simulated altitude of 300,000 feet.

—from JOURNAL OF THE FRANKLIN INSTITUTE
January, 1959

DR. B. STERNLICHT
J. C. REID, JR.
E. B. ARWAS

REVIEW OF PROPELLER SHAFT THRUST BEARINGS

THE AUTHORS

Dr. Sternlicht received the Bachelor of Science degree from Union College in 1950 and the Master of Science degree from Columbia University in 1951. He joined the General Electric Company in 1951 and was granted a leave of absence in 1953 to attend Columbia University where he received a Ph.D. in the field of Applied Mechanics in 1954. The title of his thesis was "Investigation of Translatory Fluid Whirl of Vertical Shafts." He rejoined the General Electric Company in 1954 and is considered one of the foremost lubrication specialists in the country. Dr. Sternlicht is a member of A.I.E.E., A.S.L.E. and A.S.M.E. He is chairman of the Hydrodynamics section of the A.S.M.E. Research Technical Committee and was organizing chairman of the annual 1957 A.S.M.E. session on "Stability" and also of the 1958 A.S.L.E. session on "Thermal Effects of Hydrodynamic Lubrication." He is a member of an advisory committe for the A.E.C. He has published several papers in the Transactions of the A.S.M.E. on hydrodynamics and fluid film bearings and presented a paper at the London Conference of I.M.E. in 1957 on an adiabatic solution of thrust bearings.

James C. Reid, Jr. was born on December 2, 1915 at Columbus, Ohio. He spent his youth in Detroit, Michigan, and received the degree of bachelor of Mechanical Engineering from the University of Detroit in 1940. During his years as a student he worked under a cooperative engineering work study plan at the River Rouge Plant of Ford Motor Company. In 1940 he joined the Bureau of Ships, Design Computing Section and engaged in thermodynamic and applied mechanics type calculation of marine machinery plants. During 1942 he transferred to the Propeller and Shafting Branch. In 1946 he became assistant head of the Bearings Branch. He has been the head of the Bearings Branch since 1956. He is a registered professional engineer in the District of Columbia and a member of the A.S.N.E.

Mr. Arwas received the Bachelor of Science degree in Mechanical Engineering from the University of Glasgow in 1950 and the Master of Science degree, also in Mechanical Engineering, from Columbia University, in 1951. After graduation, he was engaged in analytical and design work in heat transfer and fluid flow. Since joining the General Electric Co. in January, 1957, he has worked principally in hydrodynamics with emphasis on fluid film thrust bearings. He is a member of A.S.L.E. and an associate member of A.S.M.E.

INTRODUCTION

THRUST bearings are used in all rotating machinery to transmit the axial thrust developed by the shaft to the housing or to the foundation of the machine. One of their principal fields of application is in self-propelled ships where they serve to transmit the propeller thrust to the hull. In the United States, virtually all vessels, both merchant and combat, use tilting pad thrust bearings for this purpose.

Tilting pad thrust bearings were first produced, in the United States, by A. Kingsbury and in Eng-

land by A. G. M. Michell, around the turn of the century. Designed in accordance with principles of hydrodynamic lubrication, derived by O. Reynolds in 1886, titling pad bearings had many advantages over the previously used flat pad bearings. Thus, they could carry much higher loads and with considerably less friction and wear. They could also tolerate misalignments of the order of several minutes. The U. S. Navy quickly adopted Kingsbury tilting pad bearings and World War I saw them in general use in combat vessels. Their acceptance by the Merchant Marine followed shortly thereafter.

Today, the generally accepted range of unit loading is between 300 and 600 psi, depending on propeller speed and other factors. Future requirements, however, are expected to become increasingly severe, with regard to both load and rate of shear. A corresponding increase in thrust bearing size is undesirable because of space requirements and excessive pad and thrust collar deformations.

Notably in submarines, where space is at a high premium, future thrust bearing requirements will be particularly critical. Here too, the thrust bearing load is augmented, under deep submergence, by the hydrostatic pressure of the overhead water on the cross-section of the shaft that passes through the stern tube seal. New developments may also make it necessary for some future propeller shaft thrust bearings to operate at considerably higher speeds than at present.

The purpose of this paper is to review briefly the operating conditions of propeller shaft thrust bearings, the present status of bearing technology and to suggest areas of investigation necessary if thrust bearings are to keep up with the advancing requirements of marine propulsion machinery.

PROPELLER SHAFT THRUST BEARINGS

The principles of hydrodynamic lubrication are well known and are treated in numerous texts (Ref. 1). Later in this paper, the present status of bearing theory, the direction in which extensions of our present understanding of it are needed and the advances that have been made since the advent of high speed computers will be described.

Here, a brief review of hydrodynamic pressure generation and of the features of tilting pad thrust bearings are included, in order to facilitate later discussion of bearing performance.

In a hydrodynamic sliding bearing, load is transmitted from one to the other of two opposing faces, through a thin film of fluid, generally oil. Separation of the two surfaces is maintained by virtue of pressures which are generated in the fluid film, when one of the faces moves tangentially, relative to the other. If the two faces are parallel, as in the case of flat pad bearings, the pressures generated in the fluid film are small. If, however, a slight convergence exists, in the oil film, in the direction of motion, Figure 1, much larger pressures are generated and considerable load can be supported. Generally, the thickness of the fluid film is in the order of 10^{-3} inches while the degree of convergence is of the order of 10^{-4} inches/inch.

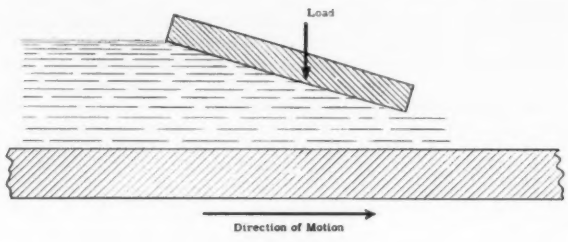


Figure 1.

The principal factors that govern the performance of the bearing are normal load, relative surface velocity, oil viscosity and bearing geometry. The load that a bearing can carry can be increased by using a higher viscosity oil; it is also greater at higher surface speeds. Here, however, limits are imposed by the onset of turbulence in the oil film which occurs at high speed and low viscosity, by excessive heat generation which raises the operating temperature and thus lowers the viscosity and by distortion of the mating parts caused by high pressures and by thermal gradients.

In the tilting pad bearing, the rotating member (or "runner") is a flat, cylindrical, steel disc that is either forged integral with the shaft or, more frequently, keyed to it. In marine applications this disc is often made up of a substantial backing collar with split facings bolted on each face, to furnish the thrust surfaces. This serves to facilitate repairs in the event of a failure. Stationary members are located on each side of the runner. Each of these consists of a number of sector shaped pads (or "shoes"), arranged in an annulus around the shaft. These shoes are generally made of steel to which is bonded a layer of tin base babbitt which forms the thrust face. Back of each shoe is a hardened steel pivot and, in operation, the individual shoes tilt and deform slightly, thus achieving the converging wedge necessary for hydrodynamic operation. Two rows of leveling plates are located between the backs of the shoes and the base ring that bears against the housing. The purpose of the leveling plates is to allow the shoes to so adjust themselves that the load is equally divided between them. They allow the bearing to accommodate small amounts of misalignment.

The double acting bearing assembly is necessary in order to transmit either ahead or astern thrusts. The ratio of ahead thrust area to astern thrust area is sometimes made equal to the ratio of design ahead thrust to design astern thrust. More generally, identical bearings are used on the ahead and astern sides for convenience in stocking spare parts and in maintenance. An axial play of several thous-

andths of an inch is maintained in order to allow the runner to bear against one set of shoes while the other face is inactive.

Generally, lubrication is provided by circulating oil under pressure through the bearing housing so that the complete bearing and runner assembly are fully submerged in the oil bath. In some cases, only the lower portion of the assembly is submerged and the oil dragged up by the runner is distributed through a system of scrapers to the upper shoes. The first of these two systems is more widely used.

OPERATING CONDITIONS OF PROPELLER SHAFT THRUST BEARINGS

By definition, under full power steady ahead the mean continuous operating condition of the thrust bearing is 100 per cent thrust at 100 per cent RPM. Certain ship maneuvers such as:

Hard rudder turn under full power
Rapid acceleration to full power ahead
Crash astern

impose different and, in some cases, more severe operating conditions on the thrust bearings.

During a hard rudder turn, the thrust on the inboard shaft increases while its RPM decreases. When such a turn is executed under full power, a particularly severe operating condition is imposed on the bearing, for it has to carry a greater thrust while its ability to carry load hydrodynamically is reduced by the decrease in shaft RPM. Figures 2 and 3 (reproduced from references 3 and 4 respectively), show typical thrust-RPM variations during such turns. Under this maneuver, the mean thrust load sometimes exceeds 140 per cent of design while the RPM drops to below 90 per cent of design. It is important also to note that these conditions are not transient but rather they last for the major part of the turn. Squeeze film action (discussed a little later) plays little part in this maneuver.

The second condition listed above, acceleration under full power ahead, is illustrated in Figure 4 (reproduced from Reference 3). Once again, the bearing is subjected to high thrust at low RPM. Here however, there are in addition large variations in both thrust and RPM before steady state conditions are reached.

It should be noted also from Figures 2, 3 and 4 that thrust fluctuations, whose double amplitude is sometimes as high as 16 per cent of design thrust, are superimposed on the mean peak thrust. These fluctuations occur sometimes at blade frequency, sometimes at shaft frequency, but in all cases their period is less than 3 seconds. Thrust measurements, however, were made using the hydraulic circuit of a Thrust Meter (Reference 2). The extent to which damping inherent in a hydraulic circuit, affects the accuracy of these transients has not yet been determined. It is quite possible that the true amplitudes of the thrust fluctuations are greater than the measured ones. This is important because these fluctuations are present throughout most of a ship's run, not just during turns or acceleration. Over a long period of time, such fluctuations, if sufficiently severe, can cause fatigue failures in the bond between the babbitt and the steel backing. There is also present the danger of cavitation of the oil film under rapid pressure variations.

The third maneuver listed above, crash astern, is illustrated in Figure 5 (reproduced from Reference

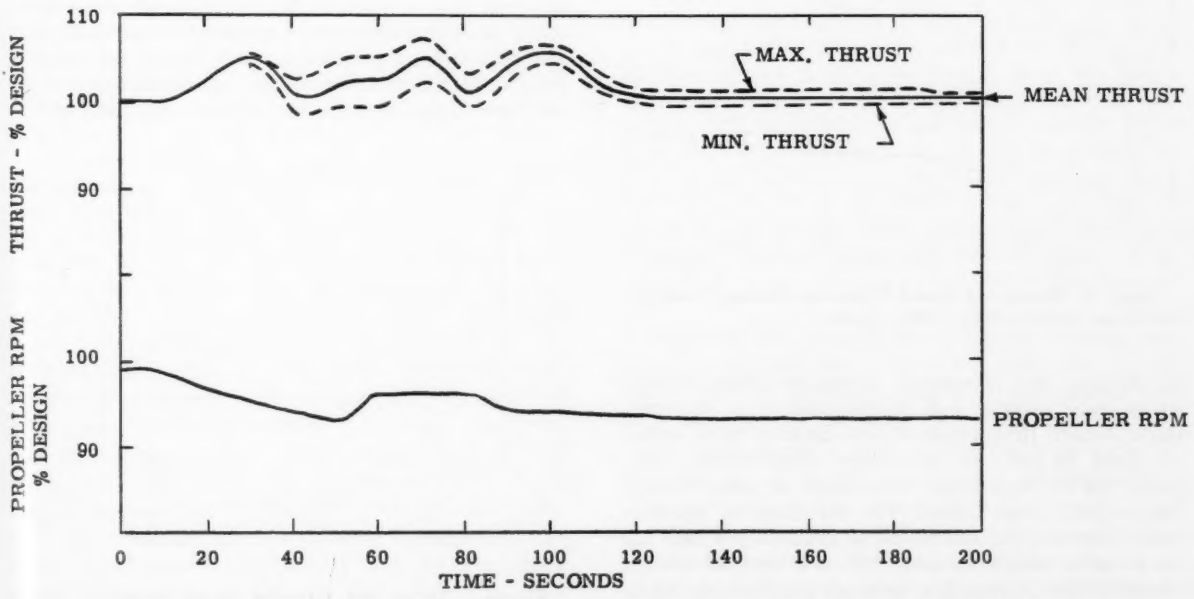


Figure 2. Thrust and Propeller Speed Variations During Turn-USS *Neosho*, 35 degree rudder.

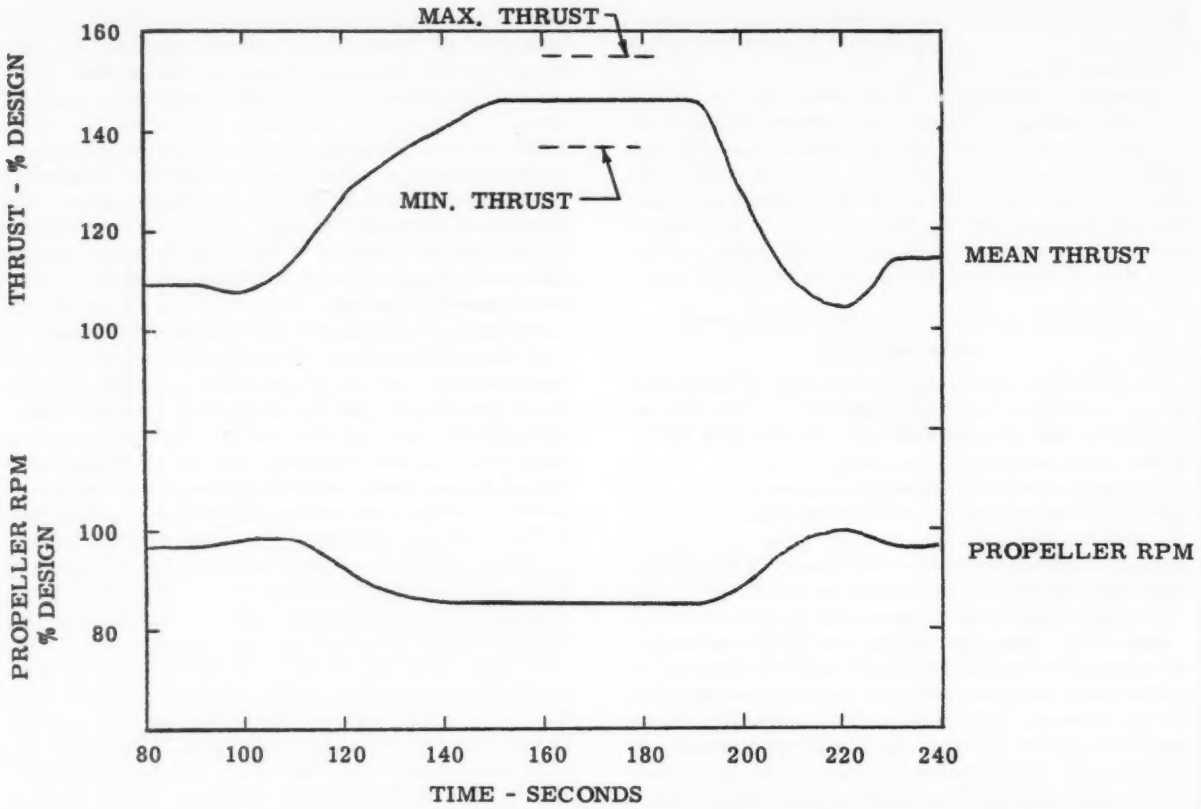


Figure 3. Thrust and Propeller Speed Variations During Turn-USS *Norfolk*, Zig-Zag Maneuvers at 35 degree rudder.

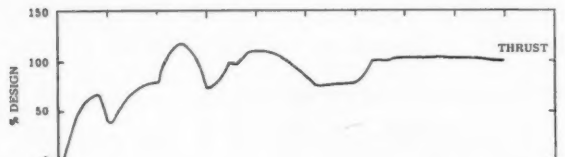


Figure 4. Thrust and Speed Variations During Acceleration from Dead-in-Water-USS *Neosho*.

5). Under this maneuver, propeller ahead thrust drops very rapidly and reaches zero in a few seconds. Astern thrust then builds up to a value equal to about 70 per cent of design ahead thrust. Propeller RPM, meanwhile, also drops to zero, though less rapidly than thrust. The direction of rotation then reverses and astern RPM gradually builds up to a value equal to about 50 per cent of design ahead RPM. During this interval, particularly when the thrust is large while RPM is at or near zero, squeeze film action prevents contact between the runner and the astern thrust bearing.

Squeeze film action (Reference 1b) arises from the fact that a viscous fluid cannot be instantaneously squeezed out from between two surfaces that are approaching each other. Because of the viscous fluid's resistance to extrusion, substantial pressures are built up that resist the mutual approach of the

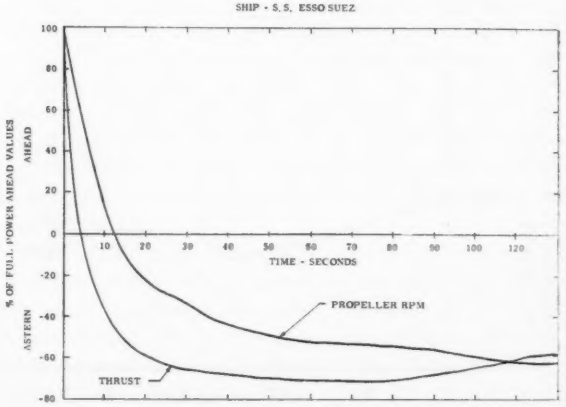


Figure 5. Thrust and Propeller Speed Variations During Crash Astern-SS *Esso Suez*.

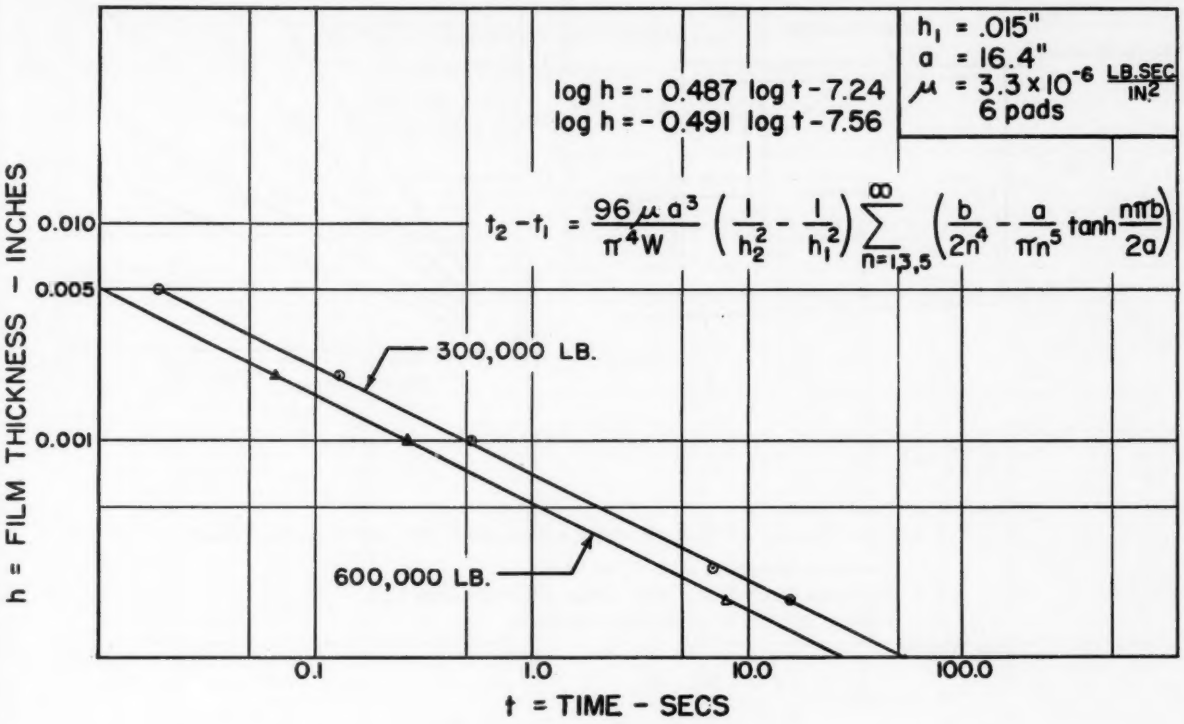


Figure 6. Film Thickness vs Time.

plates and prevent metal to metal contact for a finite interval of time. In crash astern, this is the time interval between the instant of thrust reversal and the moment when astern RPM reaches a value sufficient to generate hydrodynamic pressures whose resultant is equal to the applied load.

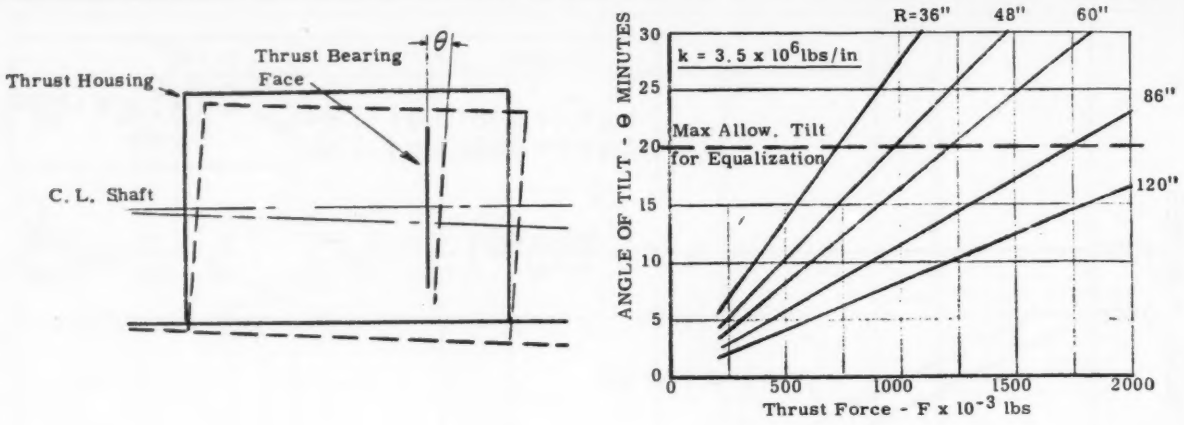
In order to estimate whether crash astern is likely to cause metal to metal contact, calculations were made for a 49″ O.D., six shoe bearing under loads of 300,000 and 600,000 lbs. The design axial end play in such a bearing is at least 0.030″. The calculation was started with the runner at mid point, that is with an initial film thickness of 0.015″. This space was taken to be filled with oil, as would be the case in a submerged bearing assembly. The results are plotted, on log-log paper, in Figure 6. It can be seen that even after 30 seconds, the separating oil film is still greater than 0.0001″. Actually this calculated film thickness is conservatively small both from the point of view of the very high loads selected and because of the assumption of zero RPM throughout the time interval. It assumes, however, that the approaching surfaces are flat, whereas any convexity of the approaching surfaces would facilitate oil leakage and accelerate metal to metal contact. A tentative conclusion can nonetheless be made, that squeeze film action will prevent crash astern thrust bearing failures.

SOME CONSIDERATIONS IN PROPELLER SHAFT THRUST BEARING DESIGN

Flexibility of Bearing Mount

Whereas in a land installation the thrust bearing mount may be made as rigid as desired, in a shipboard installation it is a function of the design and location of the bearing housing and of the ship's girder arrangement. With regard to location, the system will be flexible if the thrust bearing is located forward of the bull gear and housed integrally with the gear casing. Alternately, a much more rigid system will be achieved if the bearing is located far aft of the gear box and independently mounted (Reference 6). The bearing mount flexibility will also, of course, vary with the class of ship.

The effect of mounting flexibility on bearing performance can be visualized in the schematic of Figure 7. With a flexible foundation, the runner face is misaligned with respect to the bearing face, the angle between them being a function of the applied thrust load, of the foundation and housing spring constant and of the radius of tilt. According to measurements made at Kingsbury Machine Works, the leveling plate design allows Kingsbury bearings to retain full contact with a plane at angles of tilt



1. $\frac{F}{k} = \Delta = R\Theta$

Where: F = Thrust Force
k = Spring Constant of Thrust Housing & Foundation Referred to C. L. of Shaft
Δ = Horizontal Deflection at C. L. of Shaft
R = Radius (to Shaft C. L.) about which Thrust Bearing Tilts
Θ = Angular Tilt of Thrust Bearing (Radians)

From Which:

$$\Theta \text{ (minutes)} = \frac{F}{k \cdot R} \times \frac{180 \times 60}{\pi}$$

2. Kingsbury Bearings can Equalize at Angles of Tilt up to Θ = 20′.

Figure 7. Allowable Tilt of Kingsbury Thrust Bearings.

up to 20 minutes. The graph in Figure 7 shows, for a mount whose overall spring constant is 3.5×10^6 lbs./inch, misalignment versus thrust load, at various radii of tilt. At a thrust load of 1,750,000 lbs., for example, the theoretical limit of 20 minutes is reached when the radius of tilt is 86 inches.

Figure 8 is a table in which are itemized the spring constants of each of the components that make up the thrust bearing assembly as well as the foundation and the overall spring constants. The first column in the table shows the general range of these spring constants for large ships. The four intermediate columns show the spring constants of specific ships. Note here that two of these ships, BB55 and BB57, have the thrust bearing located forward of the bull gear and mounted integral with the gear casing. The other two ships, BB61 and CVB41, have the thrust bearing located far aft of the gear box and independently mounted, thus achieving a much stiffer system. All four of these are capital ships. Smaller ships will in general have a lower overall spring constant, that of medium size destroyers being in the order of 3 to 3.5×10^6 lbs./inch. Finally, the last column of Figure 8 shows the range of spring constants that should be built into any marine thrust bearing test machine if it is to simulate shipboard operation.

Figure 8 clearly shows that the dominant factor which decides whether the bearing mount is rigid or flexible is the foundation spring constant.

Effects of Submergence

For submarine applications, the effects of submergence on the operation of the thrust bearing must be considered. A submarine may remain deeply submerged and stationary for several hours. If the submergence is deep and the hydrostatic pressure against the propeller shaft end is not balanced at some point between it and the ahead thrust bearing, this bearing will be under high pressure while stationary. Thus, any oil film between the bearing and the runner faces will be squeezed out and dry start-up will be necessary.

Practice in land installations for large electrical rotating machinery has been to allow dry startups provided the bearing load does not exceed 400 psi. Under present marine practice, the shaft diameter is roughly equal to half the bearing O.D., thus a submergence of 2600 feet, which corresponds to a hydrostatic pressure of about 1200 psi would result in a bearing pressure of 400 psi. This indicates that a hydrostatic start should be considered whenever future submergence depths will approach 2600 feet.

Figure 8. Table of Spring Constants

(All Spring Constants Referred to Shaft Centerline)

k x 10 " #/in.

	General Range for Large Ships	Specific Ships				Recommended for a Test Machine
		B B 55 Thrust Bearing Forward of Gear Box	B B 57 Thrust Bearing Forward of Gear Box	B B 61 Thrust Bearing Aft of Gear Box	CVB 41 Thrust Bearing Aft of Gear Box	
Fluid Film	90 to 260	130	130	130	130	90 to 260
Thrust Bearing Parts	13 to 22 Std. Brgs. 22 to 36 Rigid Brgs.					13 to 22 St. Brgs. 22 to 36 Rigid Brgs.
Thrust Collar	75 to 120	10	10	20	22	120
Housing	20 to 35 Thrust Brg. Forward of Gear Box 35 to 50 Thrust Brg. Aft of Gear Box 35 to 100 Thrust Brg. Far Aft of Gear Box and Independently Mounted					50
Foundation	3 to 15 Thrust Brg. Forward of Gear Box 5 to 20 Thrust Brg. Aft of Gear Box 20 to 40 Thrust Brg. Far Aft of Gear Box and Independently Mounted	8.5	10	30	30	Variable in the Range 4 to 40
Total*	2 to 13	4.4	4.8	11.0	11.5	2.8 to 11.3 (2.8 with more flexible foundation)

* $\frac{1}{k_{TOTAL}} = \Sigma \frac{1}{k_i}$

NOTES: 1. Calculated Fluid Film Spring Constants are:

h_{min}	k x 10^{-6} #/in
0.003	90
0.002	130
0.001	260

2. Spring Constants of Thrust Bearing Parts, Collar, Housing and Foundation of Ships, per Reference 6.

It is, however, clear that at deep submergence the bearing unit loading is substantially increased. This emphasizes the need to increase load carrying capacity, by optimizing the bearing geometry.

CAUSES OF BEARING FAILURE

Records at Bureau of Ships, although not complete, show more than 30 reported thrust bearing failures since January 1954. On a single vessel, DL1, there were no fewer than 9 reported failures. Until the troubles were corrected, doubts about the ability of the thrust bearings to withstand maneuvering loads had restricted the maneuverability of the ship.

The exact cause of failure in specific cases is obscure because bearing instrumentation is not practiced. In fact the only indication of bearing performance is the outlet oil temperature indicator. This measurement gives no indication of the maximum temperature in the bearing shoes and, in fact, will indicate a failure only long after it has occurred.

There are numerous causes of bearing failure, among them:

a) Poor lubrication and, in extreme cases, bearing starvation.
b) High bearing surface temperatures which soften the babbitt lining and, in extreme cases, cause its collapse.
c) Very small thickness of the oil film which makes the bearing susceptible to dirt failures.
d) Fatigue failures of the bond between the babbitt lining and the steel backing.

e) Large distortions due to loads or thermal gradients, which adversely affect the hydrodynamic pressure generation in the oil film.

Proper precautions through good mechanical design and operating procedures will minimize the number of failures. However, it is essential to optimize the bearing geometry so that, without increasing the size of the bearings, the load can be carried at lower surface temperatures and on a thicker film. In this way the margin of safety can be increased and failures prevented.

A failed bearing shoe from DL1 is shown in Figure 9. It is typical of the series of failures encountered in this vessel. Note that the maximum wear occurred at the center of the pad, very near the pivot location. This indicates that under thermal and load distortions, the bearing pad "crowned," i.e., its working face assumed a convex shape with the apex directly above or very near the pivot. Figure 9 also shows that whereas the outer circumference of the bearing was unaffected, appreciable wiping occurred at the inner circumference. This, despite the fact that the velocity and hence the shear rate and heat generation were highest at the outside diameter of the bearing. It is clear that the radial pivot location was not the optimum one and, in order to satisfy equilibrium of moments, the pad tipped so that its inside diameter was too close to the runner.

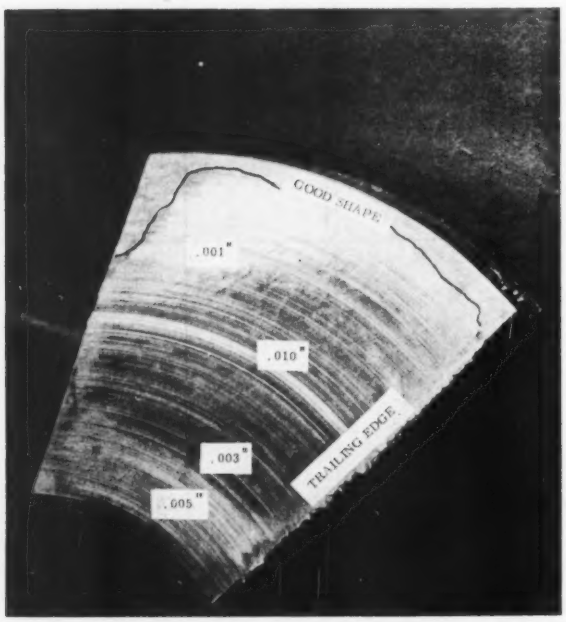


Figure 9. Failed DL1 Thrust Bearing Shoe (Blocked dimensions indicate depth of wipe).

In the preceding sections, we described briefly the operating conditions of propeller shaft thrust bearings, anticipated future needs, and some common causes of failure. We now turn to the type of program necessary in order to optimize the bearings and to the analytical and experimental tools now available.

ANALYTICAL AND EXPERIMENTAL PROGRAMS

Tests, particularly on large bearings ranging in size from 30″ to 50″ O.D., are time consuming and costly. Bearing instrumentation, essential if the tests are to be meaningful, requires precise techniques. Interpretation and analysis of test data take considerable time even if automatic data reduction devices are incorporated in the test set-up. Assurance tests which may be carried through to bearing failure require careful cleaning of the casing, flushing and filtering of the oil supply after the test.

Thus, even after the initial cost of manufacturing a test machine it is desirable to gain the necessary information for bearing design and specification with a minimum number of tests.

The functions of the analytical and experimental programs can be summarized as follows:

Experimental	Theory
1. To serve as a check between theory and practice.	1. To establish the interrelationships between the various parameters.
2. To check the validity of the boundary conditions which are employed in the analysis.	2. To guide in optimizing these parameters.
3. To evaluate problems which are too complex for existing theory.	3. To guide a test program and aid in interpretation of the results.
4. To perform assurance testing.	4. To extend applicability of test results to the full range of bearing sizes.

THRUST BEARING INSTRUMENTATION

A detailed outline of a test program falls outside the scope of this paper. In any test however, the most important dependent variables that must be measured are:

Bearing surface temperatures
Oil Film Thickness
Pad Deformation

Instrumentation for these and other measurments is available today. A summary of this instrumentation is given in Figure 10.

STATUS OF THRUST BEARING THEORY

The pioneer work in thrust bearing technology was done by Albert Kingsbury who obtained a solution of the Reynolds Equation for finite sliders using an electrical analogue. He published his results (Reference 7) in 1931. Conventional methods of calculating thrust bearing performance are still largely based on Kingsbury's findings.

An additional solution of the Reynold's Equation was obtained in 1944 using a dc network analyzer. In this solution, however, as in Kingsbury's, the oil film temperature and hence its viscosity had to be assumed constant or a known function of position.

Figure 10. Thrust Bearing Instrumentation Summary.

1. Measurement	2. Method of Measurement	3. Time Response or Frequency Response	4. Installation	5. Accessories Required	6. Accuracy	7. Remarks	
I. Bearing metal temperature	Multiple thermocouples in bearing material	½°F/sec.	Drilled holes into bearing pads	Multi-point recorders	±3°F overall	All standard equipment commercially available	
II. Runner metal temperature	Thermistors in runner metal	½°F/sec.	Drilled holes into runner	a. Indicating instrument b. Slip rings or telemeter equipment	±1.5°F overall	No recording equipment commercially available	
III. Oil supply inlet, groove, and outlet temperatures	Metal-sheathed thermocouples in fluid stream	24°F/sec.	Tack weld sheath	Same as I.	Same as I.	Same as I	s.
IV. Oil film temperatures at surface of bearing	Special thermocouple probe (see remarks)	80°F/sec.	Same as I	Continuous recording instrument	Same as II.	Sensing "bead" is flat strip 0.0001 thick, 0.004 wide, 0.10 long	t.
V. Oil film thickness	Capacitance gages	1200 cps	One in drilled hole near center of bearing, one bolted to edge of pad at each corner	a. Fielden Proximity Meter b. Oscillograph, oscilloscope, or voltmeter	±0.0001 over full range. Can be improved by good calibration technique	Some development needed for specific application	
VI. Thrust shoe movement with respect to backing ring VII. Face collar movement with respect to thrust collar VIII. Bending deformation of other static parts	Strain gage cantilever beam or proving ring transducers	2000 cps approx.	Bolt base to one part, free end of transducer contacts other part	a. Strain gage bridge-amplifier unit b. Oscillograph, oscilloscope, or indicating meter	Same as V.	Stable, linear system. All equipment commercially available. Capacitance gage system (see V) may be substituted. Slip rings or telemetry required for VII	ch
IX. Individual bearing pad thrust load	Strain gages on upper leveling plates	Natural frequency of leveling system will be limit	Apply gages directly to back of leveling plate	Same as VI-VIII	Must be analyzed for each application	System has been used by General Electric Co.	ch
X. Oil pressures in film XI. Loading cylinder hydraulic pressure	Strain gage diaphragm pressure pickups	2000 cps minimum	Drilled hole into bearing, diaphragm flush with surface	Same as VI-VIII	2% of full scale	Stable, linear system. All components are commercially available, easy to calibrate.	h

Performance predictions based on these "average viscosity" methods are generally optimistic. This has become increasingly apparent as bearing loads have been increased to keep pace with machinery requirements. Because of the average temperature assumption, the principal criterion of bearing failure was, necessarily, minimum film thickness. Thus when a bearing failure occurred that could not be attributed to external causes, such as failure of the supply line or dirt in the oil system, it was charged to a break down of the oil film and consequent metal to metal contact. In many such instances, however, the calculated minimum film thickness was considerably larger than the maximum height of surface asperities and of the largest particles of foreign matter in the oil.

This discrepancy between theory and performance led to attempts to factor the temperature variations along the oil film into the solution of the Reynolds Equation (References 8 and 9). Without the use of high speed computers, however, many simplifying assumptions had to be introduced, in order to obtain a solution. These assumptions seriously affected the applicability of the results.

Recently, however, digital computers have been applied, in the General Engineering Laboratory to the simultaneous solution of the Reynolds and the Energy Equations. This solution yields both the pressure and the temperature profiles in the bearing pads. It has explained the cause of many bearing failures by showing that maximum bearing temperatures reach the softening point of the lining material, while the film thickness is still at an apparently safe value. The procedure for obtaining such solutions is discussed in References 10 and 11 and a summary of it is given in the appendix to this paper.

Briefly then, the present status of bearing theory can be summarized as follows:

Up to 1938—Kingsbury—Slider with constant viscosity (Electrical Analogue Solution).

1953—E. Saibel—F. Osterle—One dimensional exponential film shape with exponential expression for viscosity.

1955—E. Saibel—F. Osterle—Finite sectors with exponential film shape in radial and circumferential directions.

B. Sternlicht—H. J. Sneck—Various film profiles and linear variable viscosity.

1956—B. Sternlicht—F. Maginniss—Simultaneous solution of Reynolds and Energy Equations for finite sector.

1957—B. Sternlicht—Simultaneous solution of Reynolds and Energy Equations for finite sector with arbitrary film shape and oil mixing in the grooves.

While these latest solutions have been successfully applied to the design of thrust bearings under loads of up to 1000 psi, their accuracy is limited by the amount of deformation that occurs in the bearing pads as a result of load and of thermal gradients. In the standard Kingsbury six shoe series, deformation increases roughly in proportion with the bearing size. Thus, particularly in the larger propeller shaft bearings, with a single pivot under each pad, severe crowning of the bearing pad may occur, as was suggested earlier. In order to factor this quantity into the bearing analysis, it will be necessary to program a digital computer solution of the Elasticity Equation.

Future extensions of the theory should proceed along the following lines:

Simultaneous solution of Reynolds, Energy, and Elasticity Equations.

a. To include deformations due to load.
b. To include deformations due to thermal gradients.
c. To study the effect of time on bearing deformation, i.e., creep of the bearing material.

Finally, bearing failures have occurred in the past, owing to fatigue of the bond between the bearing lining material and the steel backing. Good bonding techniques have in the past been largely dependent on the skill of the individual workman. An attempt to apply some of the fatigue theory that is available in other areas to this problem is indicated.

ANALYSIS OF A 49″ DIAMETER BEARING

As an example of the application of the theory to propeller shaft bearings, a Reynolds and Energy Equations analysis was made of a 49″ diameter, 6 shoe bearing of the type used in ships of the BB61 and CVB41 classes (Reference 12). Three additional analyses were made for bearings with the same diameter and thrust area but with 8, 10 and 11 shoes respectively. The results of these analyses are plotted in Figures 11 and 12.

Figure 11 shows that the load carried at any given film thickness increases with number of shoes until it reaches a maximum of 10 shoes. Thereafter, it starts to decrease. Thus, the optimum number of shoes for this bearing is 10 instead of 6 as called for in the present design. However, the gain in load carrying capacity between the high at 10 shoes and the present 6 shoe design appears, from Figure 11, to be only about 10 per cent.

Actually this is an incomplete picture. When the different designs are examined from the viewpoint of maximum temperature, the gain in load carrying capacity is seen to be much greater. This is shown in Figure 12. Here maximum and average bearing temperatures are plotted as functions of number of shoes. Note that at 500 psi, the maximum temperature of the 6 shoe bearing is about 250° F, whereas

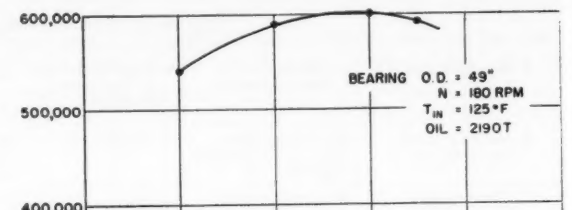


Figure 11. Total Load vs Number of Shoes as a Function of Film Thickness.

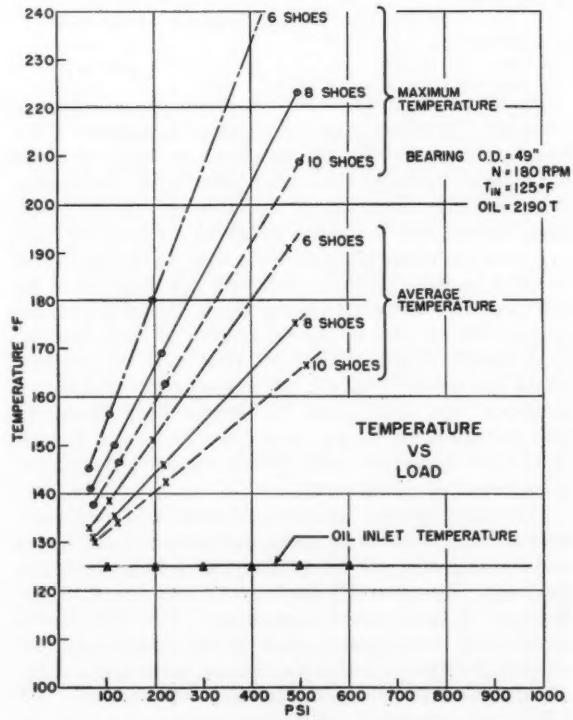


Figure 12.

that of the 10 shoe bearing is still below 210° F. The evident criterion of failure, in this bearing, is maximum temperature. In order to avoid approaching the softening temperature of the babbitt, the 6 shoe bearing has to be restricted to unit loadings of 500 to 600 psi. With only the number of shoes changed from 6 to 10, the same size bearing can be made to carry safely 700 to 800 psi.

A further factor, not considered in the analysis, is the reduced distortion that results from the smaller temperature gradients and, in the circumferential direction, from the shorter cantilevered length.

This example illustrates the fact that the available method of analysis, even without further refinement, can be used to achieve substantial improvements in bearing design.

In addition, several schemes are available to further reduce bearing maximum temperature. For example, the pad oil inlet temperature can be reduced by introducing a baffle in each oil groove that serves to minimize mixing of the hot oil discharged from the preceding pad and the cold supply oil. Another possible device is shown in Figure 13. Here helical grooves are machined in the collar backing, through which cooling oil from the sump will flow as a result of the pumping action of the runner.

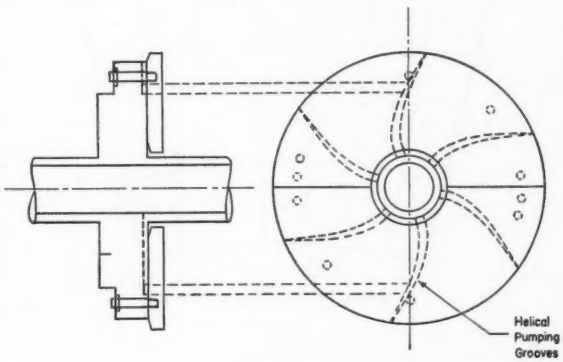


Figure 13.

DISCUSSION

The presence under certain ship maneuvers such as rapid acceleration or sharp turns, of high thrust at low RPM, as well as the number of recent failures point up a need to optimize the geometry of propeller shaft thrust bearings. The purpose of such optimization should be to increase the load carrying capacity of the bearings in order to improve their factor of safety and to reduce the risk of bearing failure. This must be achieved without any increase in size, both because of space limitations and since excessive pad distortions which accompany large size bearings adversely affect hydrodynamic pressure generation.

In this context, the analysis of a 49″ bearing, included in this paper, indicated that if the number of shoes were increased from 6 to 10, an increase in load carrying capacity of approximately 240,000 lbs. could be achieved.

Analytical and experimental techniques are now available for such optimization studies. While the equations that describe the bearing performance are quite complex, numerical solutions can be obtained relatively easily using high speed digital computers. On the experimental side, accurate instrumentation is available for the measurement of the important parameters of bearing performance, such as thrust load, speed, bearing temperature, film thickness, oil flow and pad distortion. The parallel needs of analysis and experiment in bearing studies were briefly outlined in the paper.

An important factor that must be considered when designing thrust bearings for marine applications, is the flexibility of the mount, i.e., primarily of the ship's girder system. In extreme cases, high thrust with a flexible mount, the induced misalignment between bearing and runner faces can exceed the bearing's tolerance.

Submergence effects must also be taken into account when designing thrust bearings for submarines. In the event of very deep submergence (in excess of 2600 feet), the use of hydrostatic start may have to be considered.

Recent advances in propulsion, such as high speed propellers may permit much higher speeds than are presently practical. Some redesign of propeller shaft thrust bearing systems and lubricants of different viscosity than those presently used may be required to accommodate the higher shear rates.

ACKNOWLEDGMENTS

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SYMBOLS

a =	pad length (see Figure 6)	in
b =	pad width (see Figure 6)	in
C_V =	specific heat at constant volume	in²/sec.°F
F =	thrust force	lbs
h =	film thickness	in
k =	spring constant	lbs/in
L =	radial length of pad	in
m =	number of mesh divisions in circumferential direction (see Figure 14)	
n =	number of mesh divisions in radial direction (see Figure 14)	
N =	bearing speed	rps
P =	unit loading	psi
r =	radial distance	in
R =	outer radius of bearing	in
	Also used in Figure 7 to describe radius of tilt	in
t =	time	secs.
T =	temperature	°F
Δ =	deflection (see Figure 7)	in
x,y =	coordinates	in
θ =	angular distance from inlet edge	degrees
	Also used in Figure 7 to describe angular tilt	
μ =	viscosity	lb-sec/in²
ω =	angular velocity	radians/sec

Subscripts

$_{i,j}$ = define mesh number (see Figure 14)
$_1$ = inlet quantities
$_2$ or $_0$ = outlet quantities

Bar above symbol denotes dimensionless quantity.

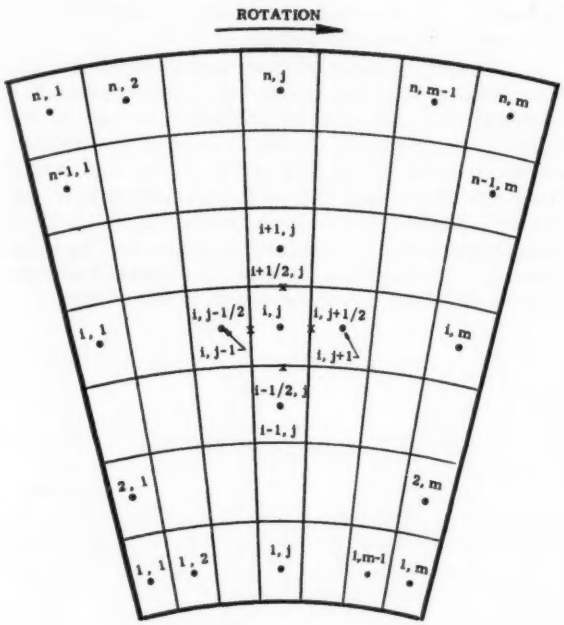

Figure 14. Mesh Notation.

APPENDIX

Thrust bearing analysis by means of simultaneous solution of the Reynolds and Energy Equations using digital computer techniques is described in References 10 and 11. Here, however, we will present the equations and briefly outline the procedure.

The Reynolds Equation in polar coordinates and expressed in dimensionless form is:

$$\frac{\partial}{\partial \bar{r}}\left(\frac{\bar{r}\bar{h}^3}{\bar{\mu}}\frac{\partial \bar{P}}{\partial \bar{r}}\right)+\frac{\partial}{\bar{r}\partial\theta}\left(\frac{\bar{h}^3}{\bar{\mu}}\frac{\partial \bar{P}}{\partial \theta}\right)-\frac{\bar{r}\partial\bar{h}}{\partial\theta}=0 \quad \ldots\ldots\ldots\ldots\ldots\ldots (1)$$

The Energy Equation, similarly expressed is:

$$\frac{\overline{\mu r^2}}{3\bar{h}}+\frac{\bar{h}^3}{\bar{\mu}}\left[\left(\frac{\partial \bar{P}}{\bar{r}\partial\theta}\right)^2+\left(\frac{\partial \bar{P}}{\partial \bar{r}}\right)^2\right]=\left[\bar{h}-\frac{\bar{h}^3}{\bar{u}r^2}\frac{\partial \bar{P}}{\partial\theta}\right]\frac{\partial \bar{T}}{\partial\theta}-\left(\frac{\bar{h}^3}{\bar{\mu}}\frac{\partial \bar{P}}{\partial \bar{r}}\right)\frac{\partial \bar{T}}{\partial \bar{r}} \quad \ldots (2)$$

The two above equations are obtained from the conventional form of the equations (Reference 13) by means of the following substitutions:

$$r = R\bar{r}$$
$$h = h_2\bar{h}$$
$$\theta = \bar{\theta}$$
$$\mu = \mu_1\bar{\mu}$$
$$P = 12\pi N'\mu_1\bar{P} \quad \text{where} \quad \left(N' = \left(\frac{R}{h_2}\right)^2 N\right) \ldots\ldots (3)$$
$$\omega = 2\pi\left(\frac{h_2}{R}\right)^2 N'$$
$$\rho C_V T = 12\pi N'\mu_1\bar{T}$$

In order to solve Equations (1) and (2) on a digital computer, the bearing pad is divided into a mesh pattern as shown in Figure 14. The two equations are then reduced to their finite difference forms which are:

(a) Reynolds Equation:

$$\bar{P}_{i,j} = \frac{\left(\frac{\overline{rh^3}}{\bar{\mu}}\Big|_{i+\frac{1}{2},j}\frac{\bar{P}_{i+1,j}}{\Delta\bar{r}^2}\right) + \left(\frac{\overline{rh^3}}{\bar{\mu}}\Big|_{i-\frac{1}{2},j}\frac{\bar{P}_{i-1,j}}{\Delta\bar{r}^2}\right) + \left(\frac{\overline{h^3}}{\bar{\mu}\bar{r}}\Big|_{i,j+\frac{1}{2}}\frac{\bar{P}_{i,j+1}}{\Delta\bar{\theta}^2}\right)}{\left(\frac{\overline{rh^3}}{\bar{\mu}}\Big|_{i+\frac{1}{2},j} + \frac{\overline{rh^3}}{\bar{\mu}}\Big|_{i-\frac{1}{2},j}\right)\frac{1}{\Delta\bar{r}^2} + \left(\frac{\overline{h^3}}{\bar{r}\mu}\Big|_{i,j+\frac{1}{2}} + \frac{\overline{h^3}}{\bar{r}\mu}\Big|_{i,j-\frac{1}{2}}\right)\frac{1}{\Delta\bar{\theta}^2}}$$
$$+ \frac{\left(\frac{\overline{h^3}}{\bar{\mu}\bar{r}}\Big|_{i,j-\frac{1}{2}}\frac{\bar{P}_{i,j-1}}{\Delta\bar{\theta}^2}\right) + r_{ij}\left(\frac{\bar{h}_{i,j-\frac{1}{2}} - \bar{h}_{i,j+\frac{1}{2}}}{\Delta\bar{\theta}}\right)}{\left(\frac{\overline{rh^3}}{\bar{\mu}}\Big|_{i+\frac{1}{2},j} + \frac{\overline{rh^3}}{\bar{\mu}}\Big|_{i-\frac{1}{2},j}\right)\frac{1}{\Delta\bar{r}^2} + \left(\frac{\overline{h^3}}{\bar{r}\mu}\Big|_{i,j+\frac{1}{2}} + \frac{\overline{h^3}}{\bar{r}\mu}\Big|_{i,j-\frac{1}{2}}\right)\frac{1}{\Delta\bar{\theta}^2}} \quad \ldots\ldots (4)$$

(b) Energy Equation:

$$\bar{T}_{i,j+1} = \frac{2\Delta\bar{\theta}\left[\frac{\overline{\mu r^2}}{3\bar{h}}\Big|_{i,j} + \frac{\overline{h^3}}{\bar{\mu}}\Big|_{i,j}\left\{\left(\frac{\bar{P}_{i,j+1}-\bar{P}_{i,j-1}}{2r\Delta\bar{\theta}}\right)^2 + \left(\frac{\bar{P}_{i+1,j}-\bar{P}_{i-1,j}}{2\Delta\bar{r}}\right)^2\right\}\right]}{\left[\bar{h}\Big|_{i,j} - \frac{\overline{h^3}}{\bar{\mu}r^2}\Big|_{i,j}\left(\frac{\bar{P}_{i,j+1}-\bar{P}_{i,j-1}}{2\bar{\triangle}\bar{\theta}}\right)\right]}$$
$$+ \frac{+\frac{\overline{h^3}}{\bar{\mu}}\Big|_{i,j}\left(\frac{\bar{P}_{i+1,j}+\bar{P}_{i-1,j}}{2\Delta\bar{r}}\right)\left(\frac{\bar{T}_{i+1,j}-\bar{T}_{i-1,j}}{2\bar{\triangle}\bar{r}}\right) +}{\left[\bar{h}\Big|_{i,j} - \frac{\overline{h^3}}{\bar{\mu}r^2}\Big|_{i,j}\left(\frac{\bar{P}_{i,j+1}-\bar{P}_{i,j-1}}{2\Delta\bar{\theta}}\right)\right]}$$
$$- \frac{\left\{\bar{h}\Big|_{i,j} - \frac{\overline{h^3}}{\bar{\mu}r^2}\Big|_{i,j}\left(\frac{\bar{P}_{i,j+1}-\bar{P}_{i,j-1}}{2\Delta\bar{\theta}}\right)\right\}\frac{\bar{T}_{i,j-1}}{2\Delta\bar{\theta}}\right\}}{\left[\bar{h}\Big|_{i,j} - \frac{\overline{h^3}}{\bar{\mu}r^2}\Big|_{i,j}\left(\frac{\bar{P}_{i,j+1}-\bar{P}_{i,j-1}}{2\Delta\bar{\theta}}\right)\right]} \quad \ldots\ldots (5)$$

Note that Equation (4) is a set of algebraic equations in which the pressure at the center of each mesh is described in terms of the pressures, viscosities and film thicknesses of the surrounding points.

Similarly, in Equation (5), the temperatures at the centers of the mesh points are described in terms of the surrounding pressures, viscosities, and film thicknesses.

After introduction of the proper boundary conditions, these two sets of equations are solved by an iterative procedure until the difference between two successive iterations is below a specified error, usually 0.1%.

In addition to the temperature and pressure profiles, the present computer program yields the load carried by each pad, the oil flow and horsepower loss and the coordinates of the center of pressure.

Official U. S. Navy Photograph

Unusual construction technique is shown in this view of two portions of the hull o f a destroyer escort, being constructed at Avondale Marine Ways, Inc.

Unusual construction technique is shown in this view of two portions of the hull of a destroyer escort, being constructed at Avondale Marine Ways, Inc.

"SHIPBUILDING AND SHIPPING RECORD"

FUEL PROBLEMS IN MERCHANT SHIPS

ACKNOWLEDGEMENT

This is an abstract of a paper presented to the Institute of Fuel in London on 14 January 1959, as published in the 22 January 1959 edition of "Shipbuilding and Shipping Record." The author is W. McClimont of the British Shipbuilding Research Association.

THERE are a number of reasons why the burning of oil fuel in marine propulsion machinery presents some problems not encountered in land practice, and the use of oil fuel in ships is therefore examined here from the viewpoint of a marine engineer in the hope that some of the problems mentioned will be of interest to fuel technologists; as far as possible the emphasis is on fuel and no more is said about engineering aspects than appears necessary. Economic circumstances are tending to make the use of residual fuel oil obligatory, and intensive efforts have been made by the manufacturers of all forms of marine main propulsion units to solve the problems associated with the burning of this residual fuel oil.

RESIDUAL FUELS

The residual fuels used in marine practice are seldom sold to a specification other than that generally known as "Bunker C grade." The version of this specification which is generally accepted is

Flash point	Minimum 150°F
Water	Maximum 1 per cent
Sediment	Maximum 0.25 per cent
Viscosity	Maximum about 6,500 seconds Redwood No. 1 at 100°F

Of these requirements, the minimum flash point has been laid down by Lloyd's, and is also needed to meet the statutory provisions of the Merchant Shipping (Construction) Rules, 1952. Normally information regarding only one of the remaining criteria is supplied to the ship's staff when fuel is taken aboard, this being viscosity. At the same time a figure for the specific gravity is also supplied.

As a general rule actual supplies have a water content well below the specified 1 per cent, and sediment of more than 0.1 per cent is rarely encountered. Although the Bunker C specification includes fuels having viscosities up to 6,500 seconds Redwood No. 1 at 100°F, the residual fuels most commonly used have a viscosity which does not exceed 3,000 to 3,500 seconds. The trend has been, however, for the general level of viscosities encountered to increase in recent years. Conradson carbon values may range up to 17 per cent and appear to be on the increase. The Conradson figure is not, in general, of significance when oil fuel is burned in a boiler, but it is generally considered inadvisable to use fuels with Conradson carbon contents much above 10 per cent in large, low-speed, marine diesel engines, and a value as low as 3 per cent has been suggested for medium-speed engines.

The ash content varies widely, the normal range being approximately 0.02 to 0.11 per cent, though a few higher values are encountered. It is not, however, the amount of ash which is of principal significance, but the nature of the constituents. Iron and silicon are troublesome because of their abrasive nature. Of the other ash constituents, the two elements which may be present in significant amounts and which may give rise to trouble are vanadium and sodium. The amounts of vanadium present appear to have increased in recent years, probably due to changes in refinery techniques.

Whatever the form in which vanadium is present, it is not amenable to removal. The complex appears always to be oil soluble and not water soluble. The sodium content is largely determined by the sea-water content. The average salt content in fuel oils of all types prior to shipment from the refineries is about 0.2 per cent, and can be ascribed to three sources, namely, water present in the crude oil, oil soluble salts present in the crude oil and, occasionally, the use of minute quantities of caustic soda during the processing of particularly corrosive crudes.

In a few cases, sulphur content is no more than 1 per cent, but it is generally higher, and the trend has been for sulphur content to increase. There are large variations in the crude oils from various sources; Middle East supplies have given marine fuel oil with sulphur contents in the range of 2½ to 3 per cent in recent years, but from most sources the sulphur content is now commonly in the region of 3½ to 4 per cent, with values of 5 per cent not unknown. There have been reports of sulphur contents as high as 8 per cent, but it has not been possible to verify these.

PROBLEMS IN BOILERS

Three main problems are encountered in boilers, namely, (1) damage to brickwork, (2) deposition and corrosion in superheaters, and (3) deposition and corrosion in air heaters.

In merchant ship boilers the main cause of brickwork wastage is spalling, caused by thermal stresses arising from rapid changes in temperature. Slag erosion is not a general problem in the direct form often encountered in Naval boilers. Gross wastage of brickwork due to slag erosion can always be associated with some mishap which has resulted in gross contamination of the fuel oil by sea water. The constituents of the fuel ash, and particularly the alkali content, are of considerable significance in the acceleration of spalling by slag penetration. On the major areas of brickwork the composition of the slag is mainly sodium and some potassium alkalis, iron oxide, calcium oxide, and a small proportion of vanadium. In these areas vanadium does not seem to be a major problem, but vanadium pentoxide slag developed in the hotter zones of the generator or superheater tube nests may attack the brickwork locally as it drips or runs down from the ends of the tubes.

Under heavy weather conditions, sea water vapor in the combustion air may supply to the boilers sodium, magnesium, potassium and calcium compounds of chlorine and other halogens and of metal oxides and sulphates. There has been evidence of the significance of sodium entering with the combustion air in cases where burner air director plates of high nickel alloy have been severely corroded.

Experience on the twin problems of deposits and corrosion in superheaters of marine boilers has been very confusing, but it would appear that ash deposition in the superheater zone is the more serious problem in current designs. Corrosion of superheater tube supports and hangers is not, however, uncommon. This is a field in which it is difficult to generalize since small differences in ash content (qualitative rather than quantitative) and firing conditions in the boiler can cause very wide differences in behavior. It should be emphasized how varied are the sources of fuel oil supplies and the fact that frequently little more than the price is known when oil is taken aboard; subsequent analysis of the behavior of boilers becomes difficult under these circumstances.

ASH FOULING

The field of ash fouling and corrosion of boilers has received considerable attention in the past 10 to 15 years. Naturally, most of the work has been concerned with land installations; simultaneously a considerable volume of work has been done on the same problems in gas turbines. Unfortunately, many people in the marine boiler field have been unduly influenced by the foregoing work and there has been a tendency for the emphasis on vanadium attack to obscure many of the more important considerations in the marine boiler application. The maximum superheater metal temperature in general use in marine boilers is probably around 950°F, while a limited number of ships are operating with metal temperatures up to 1,050°F. Most of the attention that has been paid in recent years to attack from vanadium-bearing fuels has been concerned with the temperature range 1,100°F to 1,500°F. It seems, therefore, that this work is not really relevant to the problems of the marine boiler except possibly, in some instances, to the corrosion of superheater supports. The foregoing remarks are not intended to convey the impression that the vanadium content of fuel oil is of no significance in the operation of marine boilers. Vanadium pentoxide combines with sodium, potassium and iron compounds to form substances with low melting points and these are highly destructive to high temperature boiler steelwork.

The deposits which are formed on superheater tubes are rock hard and are therefore impossible to remove with soot-blowers. In merchant ship experience they do yield, however, to water-washing and this practice is being widely adopted in merchant ships. There are disquieting reports from Naval boilers of deposits produced by fuels of higher vanadium content which are not water soluble, and the prospect of increased vanadium content cannot be other than disturbing to the marine engineer.

COMBUSTION AND FOULING

Suggestions have been made by boiler designers and others that the quality of combustion exercises

a significant control on fouling of superheaters. While good combustion is desirable for many reasons, it cannot be considered as a significant factor in reducing superheater fouling. High contents of ash-producing substances with low melting points will produce heavy fouling whatever the quality of the combustion.

Keeping air heaters free from blockages and severe corrosion has become a major item in the maintenance and repair bill, and it has been found more economical to let the funnel temperature rise and the boiler efficiency fall. By the last war, boiler efficiencies of 88 per cent had been attained, and to have this efficiency one must have funnel temperatures, even with extremely low radiation loss on the boilers, as low as 290° to 310°F. Possibly 305°F is about right. Boiler efficiency goes down roughly by 1 per cent for each 40°F rise in funnel temperature. A large number of ships are running with funnel temperatures as high as 400°F which means that boiler efficiency will be down to about 85½ per cent.

Wastage and penetration of tubes are rapid at low air heater metal temperatures, tube life being less than two years in some cases. Blockage by hard deposits of corrosion product (mostly iron sulphate) is sometimes a more serious problem than the corrosion itself, as much labor and expense is incurred in cleaning the tubes. In many instances, power-driven cutters are required to remove the deposits. This procedure is very effective, but where the tubes have been thinned by corrosion the cutters may pierce the tube walls. Build-up of deposit is rapid as the bulk of the corrosion product is great in relation to the volume of metal consumed.

The wear and fouling in diesel cylinders have received much attention in the past few years and marked improvements have been achieved. One of the earlier steps taken was to raise the temperature of the cylinder jacket which in many engines had been surprisingly low. A water outlet temperature of 165°F is now general although in some cases the water inlet temperature is still rather low at 140°F; an inlet temperature nearer 150°F would be more desirable.

Heavy duty oils containing alkaline additives were first used as cylinder lubricants for large slow-speed marine diesels quite a number of years ago, but the beneficial effect of the additives was not nearly sufficient. Up to that point the additives used had been soluble in oil, which had severely limited the choice of materials. The next step was to use additives soluble in water which could then be incorporated in the cylinder lubricant by emulsification with the oil. This opened up a whole new range of additives which have produced marked reductions in wear and fouling in cylinders. Oil solubility is a necessary property of additives for crankcase lubricants as the addition of water to crankcase oil could not be tolerated. Oil soluble additives are again coming forward with very promising results; all the major oil companies appear to be engaged in the search for new additives.

Experience of the contamination of crankcase oil when burning residual fuel soon led to the view, now generally accepted, that a well-designed, effective separation between cylinders and crankcase should be regarded as an essential condition for the use of these fuels. If this is not done effectively, bearing corrosion will result.

PROBLEMS IN GAS TURBINES

The use of gas turbines for merchant ship propulsion has been confined to the British vessel *Auris* and the American *John Sergeant*. The two factors which have retarded the development of the marine gas turbine are corrosion of the blades and deposition of combustion products in the h.p turbine; the approach to these problems differs considerably on the two ships. The problem of ash corrosion has been avoided on the *Auris* by the use of the relatively low inlet temperature of 1,200°F at which condition it has been found that corrosion has remained at an acceptable level when operating on residual fuel oils with high vanadium content. This has meant, however, an appreciable lowering of thermal efficiency, and one of the disadvantages of the gas turbine for marine propulsion is the difficulty of obtaining thermal efficiencies of the same order as the diesel engine. The turbine inlet temperature of the *John Sergeant* is 1,450°F which gives a significant boost to thermal efficiency. Arrangements have been made to combat corrosion by the removal of sodium from the fuel by a water-washing process incorporated into the fuel system followed by the addition of magnesium to inhibit vanadium attack. The fuel used so far has not warranted the use of this washing system and its service performance in coping with corrosion cannot yet be evaluated.

The water-washing process is also intended to reduce the deposition of ash on h.p. turbine blades, but here again there is no evidence of service performance. Arrangements have been made for regular cleaning; after each eight days of operation about 10 gallons of fresh water is injected into the low pressure end of the compressor, which removes the sodium and the calcium deposited by the inlet air. In addition, an abrasive compound is passed through the whole unit at similar intervals. The *Auris* set has suffered fouling by ash deposits and a number of fuel-additive tests have been done. The most effective material found so far has been crushed walnut shells.

It seems clear that the search for fuel additives to prevent fouling and corrosion will go on, as the future of the gas turbine as a marine propulsion unit depends largely on the achievement of an increase in thermal efficiency by the use of higher temperatures.

The third surface ship to operate the Navy's missile testing program, and the first one to participate in the POLARIS launching program, has recently been commissioned. Converted from a Mariner hull, USS OBSERVATION ISLAND is particularly noteworthy because of the precision navigational gear, which features automatic celestial tracking equipment for providing continuous navigational information. The ship is equipped with extensive workshops and instrumentation which enable it to give full support to weapon system research and development. It is planned that POLARIS missiles will be launched from the ship as a part of the missile development program.

—from BUREAU OF SHIPS JOURNAL, March, 1959

An RCA image converter has been successfully used to photograph stars electronically. A large step toward the goal of electronically amplifying starlight, the converter has been successfully used to photograph 18th magnitude stars through the U. S. Naval Observatory's 40-inch reflector with a two minute exposure. The converter is a six-inch long tube which receives starlight from the eye-piece of a telescope at one end, intensifies it by electronic means, and displays the image on a luminous surface at the other end where it may be photographed.

—from SCIENCE NEWS LETTER, March 7, 1959

Details of a previously classified navigation system for submarines have been recently released by the Navy. The system, called SCAR for Submarine Celestial Altitude Recorder, makes it possible for submarines to take fixes while submerged, sightings being accurately taken through the periscope. The Sperry developed system comrises a highly accurate altitude measuring and recording device incorporated in the periscope. The sight is taken by pressing a button when the optical image is aligned. The altitude and time are automatically recorded. The system was used successfully in the recent record-breaking submerged voyages of NAUTILUS, SEAWOLF and SKATE.

—from NAVY PRESS RELEASE, March 11, 1959

ENSIGN RICHARD L. NELSON, USNR

ELECTRONIC COMPUTERS MANAGEMENT APPLICATION

THE AUTHOR

is a native of Mansfield, Ohio, born 18 November 1934. He partially financed his college career by means of funds he obtained from the sale of a device he designed and built for automatically controlling the flow of untreated stone in asphalt plants. During his college years, while employed by the Hartman Electrical Manufacturing Company, he conducted research into the operation of electrical contacts, which data provided the groundwork for his thesis for the degree of Electrical Engineer, which he received from the University of Cincinnati in June, 1957. Mr. Nelson received first prize in the Cincinnati Section in the American Institute of Electrical Engineers prize paper competition for a paper on Contact Arcing Phenomenon, a few weeks prior to entering Officer Candidate School, Newport, where he was commissioned Ensign, U.S.N.R. on October 31, 1957. He is registered as an Engineer-in-Training in the State of Ohio, and is an Associate Member of the A.I.E.E. He is presently stationed at the Norfolk Naval Shipyard where he is engaged in developing computer program material for management application.

INTRODUCTION

NEARLY everybody is inclined to associate the word computer with abstract mathematical formulae. To most of us, the very word compute brings to mind the scientist, writing hieroglyphics on a large blackboard, speaking a language foreign to the uninitiated. In a lesser sense, we may compute the cost of repairs to our house, or more colloquially, "figure" our bank balance. Probably none of us would apply the term to the mental calculations we become embroiled in when trying to decide which item, or items we shall select when confronted with a lengthy menu while out for dinner. When it is explained that the machines we have come to call "digital computers" can easily be applied to all of the above situations (assuming you can define your tastes in food and pocketbook limitations in a few thousand words) it is usually received with a great deal of scepticism.

There are probably several reasons for the common misconceptions concerning the function and capacity of such devices. At least one of them is our language, and the fact that these machines defy simple explanation. In a sense, we are at grips with the same problem which the interpreter at the United Nations faces while attempting to give a word for word account of what the bald headed, foreign orator in the front row is saying. There just is no single word which adequately describes the complex nature of these devices. To say merely that they compute, in the definitive sense is as limited as saying that a bank is a place to save money. Both statements are true as far as they go, but in the case of the bank, you have failed to mention any of the rest of their multitudinous functions. So it is with the digital computer.

Actually, these machines are capable of performing a variety of logical operations, including their arithmetic functions. In fact, arithmetically, the ma-

chine is able to do only two things: add and subtract! Were it not for the logical abilities of the device, it could not even perform the intricate operations of higher mathematics. The fine press releases these machines have received, regarding their ability to perform fantastic mathematic feats undoubtedly earned them the respect of all. Unfortunately, it also "type cast" them, and was probably largely responsible for earning them the reputation of being installed in ivory towers, and out of reach of all except the engineering branches of industry.

Since the field of mathematics constitutes a system of logic in itself which can, theoretically at least, be applied to anything which can be counted, measured, or represented by some numeric factor, there is little wonder why the machine was so readily applied in the mathematical field. When it is more generally realized that these machines are capable of solving nearly any problem in which the solution is dependent upon the logical handling of information, regulated by boundary conditions, then we may begin to exploit these devices to the full extent of their capacity.

THE COMPUTER AND MANAGEMENT

Although no single system, such as exists in the field of mathematics, has been developed to handle them as easily, our business and productive operations are, or should be, based on some logical motivation. While it is not always clear cut, at any particular time there is usually some method, operation, or program which based upon present knowledge and past experience should yield optimum results. Mentally, we attempt to determine the conditions productive of this best result by logical reasoning, i.e., by exercising judgement. In arriving at this conclusion, we have drawn upon our knowledge of the situation and, if available, our past experience in dealing with similar situations.

In solving problems which are not directly reducible to mathematical formulae, the digital computer functions in an analogous manner. Within its memory capacity, the machine may examine hundreds of alternative approaches to a problem in the light of thousands of pertinent knowledge factors, and yield a result of the best course of action in a split second. The same critical examination by management personnel may have occupied a major portion of their time for days, and the degree of their subjective influence upon the factors involved represents a variable not readily visible in the answer. The computer exhibits the ability to decide logically upon the most objectively appealing of a group of alternate choices, on the basis of the information it has available. Its choice is completely objective, and its "accuracy" of selection is dependent only upon the accuracy of the information which it has available. A computer, properly programmed, is also able to show the effect of a change in the data with which it works quite readily. In the engineering field the ability of another type of computer, the analog computer, to continuously display the effect of changes in its input data makes it valuable in complex systems analysis. (For this type of computer, unlike the digital machine, the input-output relationship must be expressed in mathematical form.)

To some small extent, the manner in which a computer may be employed in the solution of managerial problems, and how this is related to the mathematical aspects of the device has been explored. To merely use the computer to perform that work which has been handled by managerial assistants in the past is not usually sufficient to justify its use. The factor of the speed with which the device is able to generate data useful to management is advantageous, and often very important. Probably the biggest single factor in enumerating the points for justification of a computer installation for management application is that it will be able to supply information heretofore unobtainable without many hundreds of additional managerial personnel. This would, obviously, produce in turn a top-heavy system. Overall, the ability of a well integrated computer system to yield rapid results, and produce sufficient and reliable information for tight production control in a large industrial complex, should enable its financial justification.

THE FIELD OF INDUSTRIAL AND MANAGERIAL APPLICATION

Examples of specific applications of computers to the solution of logical, but not necessarily arithmetical, problems may be drawn from nearly every field of industrial activity. The relatively new field of operations research frequently makes use of the logical abilities of the digital machine in the solution of problems and studies well removed from the arithmetic field.

In attempting to apply the computer to management problems, it has usually been found necessary to carefully scrutinize existing control procedures. Published experience along this line indicates that such an examination is a profitable idea in itself, and often results in timely alterations of the management control system, and is productive of more profitable operation, computer applications notwithstanding.

In spite of the diversity of past and present application of these machines, today's computers remains technically ahead of management's capacity to exploit them.* This means that while the computer has the ability to solve many of management's problems, industry in general has been hard put to express their problems in the logical form demanded by the computer. In short, the apparent lag, if it exists, in the efficient, industrial utilization

* Peter James, "9 Guideposts in Selecting a Business data Processor," *Control Engineering*, January 1958.

of the computer is, for the most part, due to the inability of management to adequately define their problems, and properly express themselves in language useful to the computer. As a direct result of this difficulty, large numbers of successful management consultants now specialize in assisting top manufacturing management in analyzing their manufacturing operations and management problems, with an eye toward computer application.

Convair Aircraft Corporation* uses a digital computer to determine their workload distribution daily, and to control the flow of work throughout the plant. Much of the work handled in this manner is of a non-repetitive nature, and the problem is complicated by the fact that the system must keep track of 600,000 detail parts as they travel between fabricating areas to assembly areas. By knowing the daily workload distribution, and the status of fabrication and assembly areas, they are able to more effectively utilize their manpower capabilities, and avoid work pile-ups in some production areas, while others remain slack. The net result is a more efficient utilization of manpower, and a more regular, rapid flow of work through the various departments within the plant. The computer also provides a printout, in condensed form, of this work status, indicating those areas deserving of special attention, due to a deficiency or excess of work. This report is available at the beginning of each working day, thus allowing management the opportunity to take timely, corrective action in the exceptional areas with a minimum loss of time. Under former procedures, hundreds of mandays of productive effort may have been lost before the implications of a condition came to light, were analyzed and acted upon. Officials of the company expect the system to be productive of historical data, which will form a firm basis for shop scheduling and load forecasting; in the past, such factors were largely a matter of experience.

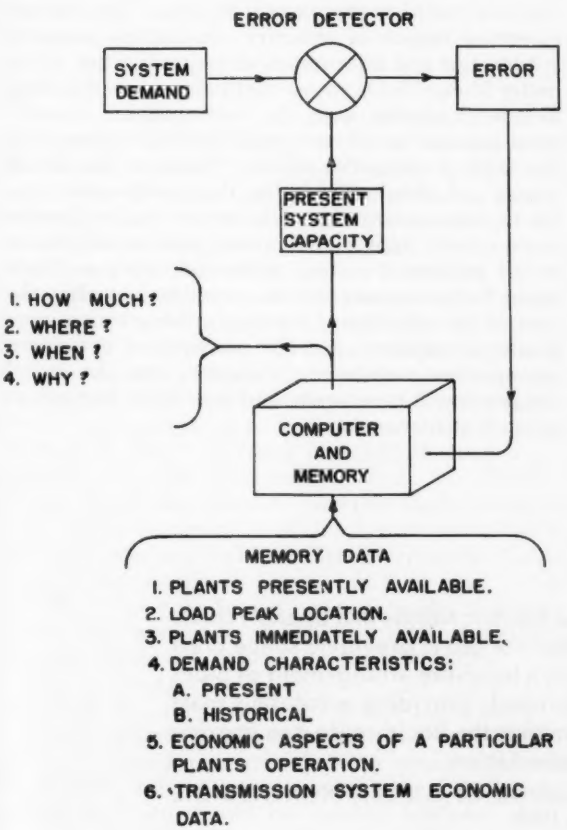

FIGURE 1

One power system, covering several of the southern states, uses an analog computer to examine their workload requirements (power demand) in various locales, and determine the most efficient and economical utilization scheme for the units available to be "put on the line." This computer, known as the "Early Bird,"* generates data on a semi-continuous basis, upon variations in power demand, both anticipated and real. It is not only being used to analyze data on power requirements, and determine optimum utilization reports for action by management, but actually serves a major function of management, since it analyzes its own generated data, on workload demand, and tells the dispatcher exactly which units should be energized. The optimum utilization possible with this system has resulted in financial savings for the power complex greatly exceeding their anticipated value.

Each of the above examples is diversified in nature of application. In spite of this diversity, however, the general approach to their solution was undoubtedly similar. The end result desired in each case was the efficient control of some productive effort. Initially, the factors entering into the problem were tabulated and analyzed. A closer examination of the power complex may reveal some of the aspects of the problem.

Forty-one different generating units in fourteen separate plants are available for use on the power system. The block diagram shown in figure 1 lends itself well to our purposes, since it can be used to illustrate some of the factors involved in the problem. The "error detector" is the device which lets us know a problem is at hand. In this case, the line frequency may be examined to determine the overall system behavior. Individual voltage levels may show where present equipment overloads or underages are present.

The factors desired in the solution are:

1. How many units are to be added or removed from the line?
2. Which units are to be added or removed? "i.e." where are the alterations to take place, in order that the altered system will be operating in

* "Keeping Tabs on Production," *Control Engineering*, January 1958.

* The *New York Times*, Business Section, July 6, 1958; Page F-1.

the most economical fashion for present system characteristics?

3. Considering the characteristics of the load variation, when will it become necessary to alter the system? This relies on the ability of the machine to predict future events on the basis of the information it has at hand. Due to the inertial aspects of a turbine unit, this is particularly important when a device is to be brought up to speed for "putting on the line."

Factors which must be considered in reaching the solution, aside from those mentioned above, might be:

1. What plants are available?
2. What are the economic factors concerning a particular plant's operation, as, for example, the cost of fuel in a certain locale?
3. What is known concerning the historical demand characteristics in particular regions?

This example is oversimplified, of course, but outlining the problem in a similar form was probably one of the initial steps in the overall solution, and these are, indeed, some of the factors which the control systems designer had to take into account in developing the computer program. This is all part of the process of defining the problem. In the back of our mind, throughout the process, is the objective we are trying to achieve; meeting the power requirements in the most efficient and economical manner. In most managerial computer applications, this is the fundamental target. Even if the machine is used only to provide reports for management review, it is usually anticipated that these reports will provide that which was lacking in any former system, and ultimately bring about more efficient management control. How well the object is achieved can usually be measured in terms of the overall increase in *productive* effort available to be expended in useful work. The success encountered in this objective is a function of how well the problem and the factors involved in it were evaluated, how well the system has been integrated into the overall productive effort, and its degree of application.

CONCLUSION

It is difficult in a few pages to accomplish a great deal in the way of shedding light on the wealth of material involved in any application of a computer system. Any such attempt is usually about as fruitful as trying to learn about a university by taking a Sunday afternoon stroll through its campus. It is hoped that some of the ideas presented here will stimulate thought on the manner in which a computer may be of benefit to management. The device itself is a highly developed tool, and the fact that it can be valuable to management has been demonstrated by those industries quick to realize its potential and use it to advantage.

In the last fifteen years, management has had to become more technically minded in order to reap the benefits of a growing technology. This is evidenced by the growing tendency of management to increase its ranks from among technically competent personnel within industry, as well as making technical advisers available for consultation. The last ten years have seen a phenomenal growth in sciences useful to the executive rather than the engineering branch of industry. Operations research, cybernetics and information theory are a few of the better known fields whose methods and studies deal, to a large extent, with the factors which management has had to contend with since the inception of the modern industrial system. There is little doubt among industrial officials but that as the ideas useful to management latent in these studies become more widely appreciated, there will be alterations in the industrial system as revolutionary as Frederick Taylor's study was to produce just after the turn of the century. Considering the growing complexity of industry, and the evolution of these new management techniques, it appears that the digital computer will necessarily and inevitably become an adjunct of management.

Mr. Robert P. Haviland of the General Electric Missile and Space Vehicle Department has patented a liquid flywheel for use in providing space craft with a stable orientation. The apparatus is a hoop-like arrangement of pipes through which a quantity of fluid is pumped, providing a rotating mass resulting in gyroscopic properties. Changing the liquid route can provide processing forces to obtain desired re-orientation.

—from JOURNAL OF THE FRANKLIN INSTITUTE
January, 1959

COMMANDER ROBERT S. BURPO, JR., USNR

A NOTE REGARDING THE INTRODUCTION OF STEEL IN THE U. S. NAVY

THE AUTHOR

is a Reserve Engineering Duty officer on extended active duty, recently assigned to the David Taylor Model Basin as High Speed Dynamics Program Officer, and presently on duty in the Bureau of Ships. Other recent duties: student at the U.S. Naval Postgraduate School, Monterey, California; Project Officer at the Material Laboratory, New York Naval Shipyard.

WHILE reading the recently published *Picture History of the U. S. Navy* (1) some comments in the section entitled "New Navy" aroused my curiosity. The decadence of the United States Navy following the War Between the States was terminated by the construction of the ABCD warships; these were the first ships of the "New Navy." *Atlanta, Boston, Chicago* and *Dolphin* embodied many mechanical innovations and their hulls were constructed of steel plates. These four ships comprised the Squadron of Evolution* which seems to be analogous to our present Operational Development Force. And, according to the authors of the *Picture History of the U. S. Navy*, the ABCD ships were planned ". . . as iron-hullers. But Lt. Robley Evans, Navy metallurgist, inserted steel specification [*in the construction contract with John Roach and Sons, the shipbuilder*]. Roach protested; Navy Department transfered Evans, but kept steel." The details of this episode involving a change in 'specs' might be of interest to an ED officer, so I visited the National Archives seeking the complete story.

After a study of the documents on file in the Archives (2), (3), (4), (5), the tale takes form.

In an attempt to rejuvenate the Navy in 1881, an Advisory Board recommended the construction of 38 modern ships. Of this number, the Board suggested that two should be 15-knot, 5,873-ton steel cruisers; six should be 14-knot, 4,560-ton steel cruisers; ten should be 13-knot, 3,043-ton steel cruisers; and twenty should be 10-knot, 793-ton wooden cruisers. Commander Robley Evans was a member of the Board. It is interesting to note that when the Board voted on steel for the hull material for the 14-knot cruisers the President (Admiral Rodgers), the two Chief Engineers (B. F. Isherwood and C. A. Loring) and the three Naval Constructors (John Lenthall, T. D. Wilson and Philip Hichborn) on the Board cast the negative votes. Steel was recommended, however, by an 8 to 6 vote. When voting for steel as the constructional material for the 13-knot ships, the "Noes" were cast by four members: the two Chief Engineers and two Naval Constructors. The count of the vote on the 15-knot vessels was not recorded in the minutes of the Advisory Board (2). Apparently there was considerable difference of opinion as to the advisability of building these proposed ships of steel.

The following year (1882) a second Advisory Board was convened to establish detailed specifications for the first four ships to be constructed under this modernization program. The second Board was made up of the following members: Rear Admiral Robert W. Shufeldt, President; Mr. Henry Steers, Naval Architect; Mr. Miers Coryell, Marine Engineer; Chief Engineer Alexander Henderson; Commander John A. Howell; Lieutenant Edward W. Very and Naval Constructor Frank L. Fernald. There was no unanimity of opinion regarding the use of steel amongst the members of this board, either. The specifications that resulted from the deliberations of the second Board authorized the

* Also called the White Squadron (from the color of the paint worn by the ships) but not to be confused with the Great White Fleet that circumnavigated the globe in 1907-1909.

ABCD ships to have hulls of "steel of domestic manufacture having as near as may be a tensile strength of not less than 60,000 psi, and a ductility in 8 inches of not less than 25 per cent." A minority report dissenting from the selection of steel was filed and it consisted of six pages of hand-written argument describing the advantages of iron over the then obtainable grade of steel (4).

From the record, it seems that the decision to specify steel was argued during the deliberations of the two Advisory Boards and was made firm before any bids were requested or contracts awarded.

The following note ended my search for evidence of a Roach versus Evans difference: "These vessels are to be built in accordance with all the requirements of the Acts of Congress of August 5, 1882 and March 3, 1883.* They are to be constructed of steel of domestic manufacture . . ." (5)

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4. Report of the Secretary of the Navy, Vol. I, 1883 (Government Printing Office, Washington, 1883).
5. Memoranda Concerning Construction of New Vessels 1882-87 (Navy Department).

*The Acts authorizing the construction of the ABCD ships.

Modernization of the SS Constitution and SS Independence is nearly a seven million dollar improvement program. This undertaking involves enlarging of the passenger carrying capacity of each ship to 1100 passengers by adding 50 new cabins and suites. In an unusual structural procedure, a portion of the existing superstructure, measuring 42 feet long, 61 feet wide and 17 feet high, was cut loose and relocated forward and one deck higher. The 150 ton unit was transferred a distance of 21 feet on an inclined plane, formed of 18 inch I-beams, using geared chain-falls. This portion of the superstructure included the pilot house; bridge wings; chart house; gyro compass; radio and damage control rooms; and several officer's quarters. Completion dates for the modernization, taking place at the Newport News Shipbuilding and Dry Dock Company, were February 20 for the Constitution and April 25 for the Independence.

—from MARITIME REPORTER
February 15, 1959

The Army Package Power Reactor (APPR-1) has accumulated over a year of operating experience since first going on the line in November 1957. The installation is designed for a thermal output of 10,000 KW and an electrical output of 1855 KW. In a 700 hour acceptance test, the reactor was required to operate at various loads between 10% and 100% of capacity with a maximum permissible shutdown time of 40 hours. The test was completed with a little less than 7½ hours for shutdown. The reactor demonstrated a particularly good capability of following full range load changes without a scram or control rod adjustment. Several relatively minor operating difficulties have been encountered during the period since acceptance, but they have been overcome without extensive modification. The greatest continuing problem has been reliability of instrumentation centering around vacuum tube performance.

—from NUCLEONICS
February, 1959

STEAM GENERATOR DESIGN TRENDS

ACKNOWLEDGEMENT

This reprint consists of four papers presented at the National Power Conference of the ASME and AIEE, in Boston, Mass., 28 September-1 October, 1958. The authors are: Part I—E. M. Powell and J. I. Argersinger, of Combustion Engineering, Inc.; Part II—D. R. Wilson, of The Babcock and Wilcox Company; Part III—D. B. Stewart, of Foster Wheeler Corporation; Part IV —C. F. Hawley, of Riley Stoker Corporation. Reprinted from the November 1958 edition of "Mechanical Engineering."

I—A Preview of Tomorrow's Boilers

AN EXTRAPOLATION of successful operating experience would indicate that the boiler of tomorrow will generate 3,500,000 lb. of steam per hour to supply a 500,000-kw unit operating at 3500 p.s.i. and 1050°F with double reheat at 1050°F. Economics demands the use of such high-capacity units to meet the doubling of the electric power load which has taken place every eight or ten years. It was predicted in 1956 that a unit size of 500 mw would be achieved by 1960. Actually, this has occurred two years ahead of schedule. The Tennessee Valley Authority now has on order a boiler of 3,700,000-lb. per hour capacity to serve a 500-mw turbine generator.

Likewise, economics demands equipment of increased efficiency and plant cycles with lower heat rates to offset the rising price of fuel. In general, this means the use of higher steam pressures and temperatures.

SUB AND SUPERCRITICAL PRESSURES

Operating pressures have increased from about 200 p.s.i. at the beginning of the century to above 2000 p.s.i. in 1957, with the standard or generally accepted maximum at 2400 p.s.i. at the turbine throttle. We are face to face with the critical pressure, and the next logical step, the proper engineering step, is to go above it.

What is critical pressure? What does it mean to generate steam above critical pressure? In an enthalpy diagram, the saturated-water line and saturated-steam line meet at the critical point of approximately 3206 p.s.i. and 705°F.

In a boiler, heat is added at essentially constant pressure, and the temperature rises until the saturation temperature for the existing pressure is reached. As further heat is added, there is no change in temperature, but the water boils and steam bubbles are formed. The volume of the mixture of steam and water increases rapidly, especially at the lower pressures, until the water has been evaporated, and dry steam—still at saturation temperature—remains. Further addition of heat brings us into the superheat region, with the temperature and volume increasing in a manner that approaches the perfect gas laws.

At some pressure above critical, say 4000 p.s.i., water, which is practically incompressible, will have

almost the same specific volume as water of the same temperature at any other pressure. It will also have about the same heat content. As we add heat, the temperature rises, about 1 degree for each B.t.u. per lb. of water, just as it does below critical pressure, and the volume increases very little. As the temperature gets above 700°F, the rate of increase in temperature becomes slower, but the rate of increase in specific volume becomes more rapid. We are getting into the transition zone. There is no boiling, or separation of steam bubbles. There is no zone in which the temperature remains constant while we add latent heat. There is no condition which we can identify as marking the boundary between the liquid and vapor phases. However, by the time we get somewhat over 800°F, the characteristics of the fluid are completely analogous to superheated steam below critical pressure. Certainly for all engineering purposes it can be considered steam.

A little consideration of this process of forming steam above critical pressure leads quickly to the conclusion that conventional boilers are not suitable. A boiler of the once-through type is required.

BOILER CIRCULATION TYPES

The basic differences between types of circulation in boilers are simply illustrated in Figure 1. In a natural-circulation boiler feedwater is pumped through the economizer into the boiler drum. It passes through downtake pipes or tubes into the bottom of the heated evaporating surface. This is usually the water wall in modern designs. The addition of heat causes boiling, and the decreased density of the boiling mixture creates circulation. The mixture of steam and water returns to the drum. Steam is separated from the water and is forced through the superheater to the outlet of the unit.

We might call this yesterday's boiler, although this does not imply that it has outlived its usefulness for many utility and industrial plants at moderate pressure levels.

Since the circulation depends on the difference in density between steam and water, the rising trend in operating pressure gave the designer an increasingly difficult problem. An answer is provided by the controlled-circulation design which has been widely applied during the past 10 years in the pressure range above 2000 p.s.i. but below critical pressure. Pumps are provided to produce positive circulation, independent of heat absorption or relative density of steam and water mixture. Orifices are used to distribute water to parallel circuits of varying resistance in proportion to their requirements.

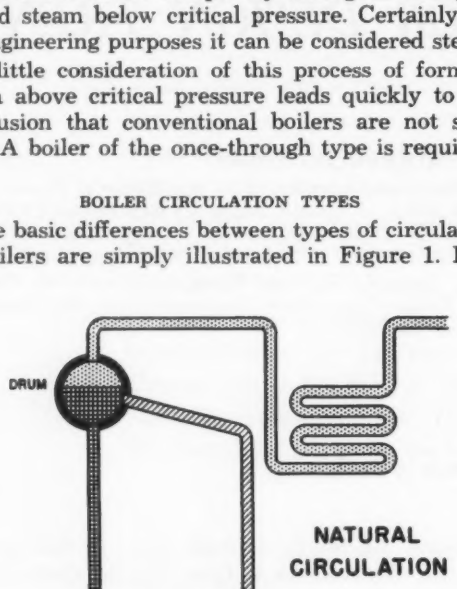

Figure 1a.

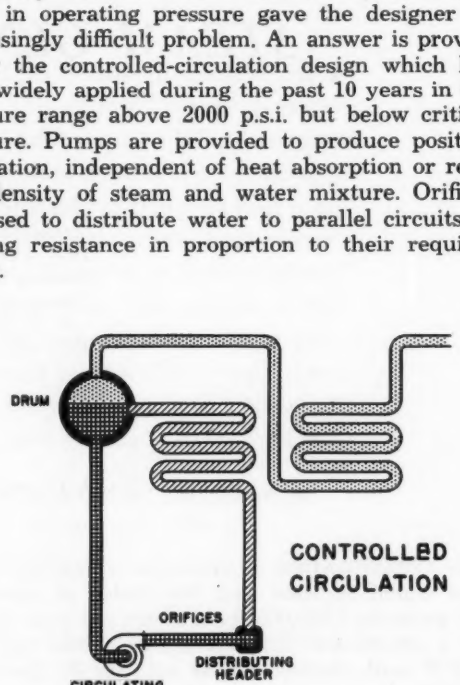

Figure 1b.

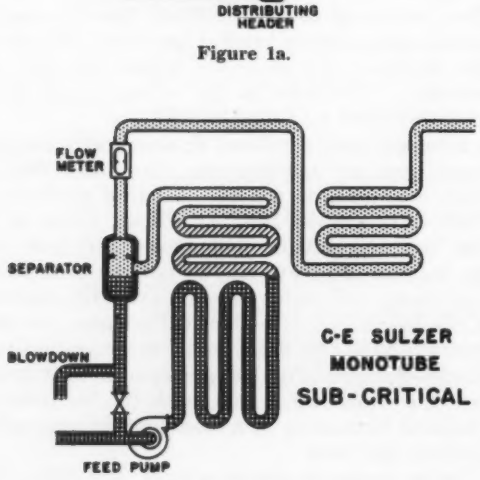

Figure 1c.

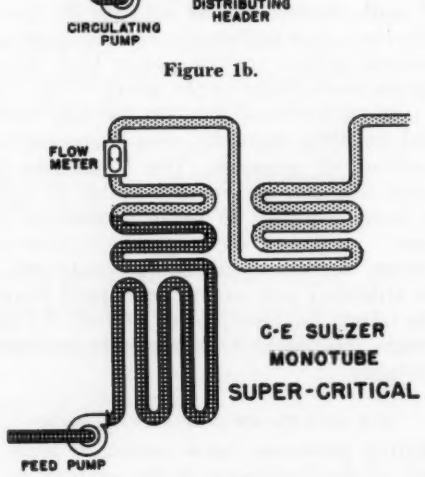

Figure 1d.

Over 100 units of this type have been ordered up to the end of 1957 with a total capacity of almost 130,000,000 lb. per hour representing over 23 per cent of the industry purchase in sizes over 50,000 kw.

MONOTUBE BOILER—SUBCRITICAL PRESSURE

The monotube boiler is a once-through boiler which can be applied either above or below the critical pressure. The author's company now has six steam-generating units of the monotube design in various stages of construction or in operation, representing a combined capability of nearly 1,500,000 kw. In the design for subcritical pressure there is no recirculation of water within the unit. The fluid passes in sequence through the economizer, the evaporating portion, and the various sections of the superheater.

In a large-capacity unit, tubes in parallel are necessary to handle the flow of water and steam. The stable distribution of flow between these parallel circuits is assured by several features of the monotube-boiler design. This factor of stability is most critical in the evaporating section where a mixture of two phases must be maintained. The continuity of tubes throughout the evaporating section eliminates the problem of handling a mixture of water and steam in a header and distributing both phases uniformly into a large number of tubes. It is this feature of continuous tubes which gives rise to the name "monotube."

A second design feature is that all the tubes pass several times in parallel through all parts of the furnace, so that each receives the same total amount of heat, thus minimizing the effect of any local disturbance, such as burner adjustment or soot-blower operation.

An orifice is located at the inlet of each water-wall tube to provide a fixed resistance, to improve stability, and to control distribution in the same manner as in a controlled-circulation unit. Finally, an adjustable resistance in the form of a seatless valve is included at the inlet to each small group of tubes, permitting adjustment for unavoidable minor differences due to design or fabrication.

Another feature of the monotube design for subcritical pressure is the boiler-water separator. The feedwater is controlled to maintain a small amount of moisture in the steam at the water-wall outlet. The moisture is removed in the separator, carrying with it, in concentrated form, a large portion of any impurities in the feedwater. The discharge from the water separator can be piped partially to continuous blowdown, and the remainder returned to the cycle in the high-pressure feedwater heaters. High purity of feedwater is a requirement of all once-through boilers, especially if long-time continuous operation is to be achieved. With a water separator the requirements are slightly less critical, because chemical blowdown can be utilized. Also the evaporative circuits can be washed or flushed during operation, by temporarily increasing the feedwater flow slightly above normal requirements.

The water separator also maintains the steam at the superheater inlet at a constant condition, dry and saturated, making regulation of the outlet temperature more accurate and uniform.

Finally, the water separator makes it possible to restart the boiler quickly when hot, without quenching the hot superheater metal, and with a minimum change in superheater-outlet temperature. The firing rate required is less, and more uniform temperatures are produced throughout the steam-generating unit, lessening expansion problems.

The monotube boiler, at subcritical pressure, is well adapted to the use of radiant-superheat surface, with the superheater protected by positive flow during all conditions. The use of high steam temperatures and reheat temperatures make radiant superheaters more attractive. The monotube design eliminates the large boiler drum and minimizes the quantity of water-wall headers and connecting piping, particularly as compared with natural-circulation units.

The monotube boiler is extremely flexible in operation. It is ideally suited to quick starting, and it can be operated over a wide load range. By taking advantage of the water separator, the steam output can be reduced to 10 per cent or even lower, while still maintaining adequate flow in the furnace tubes.

MONOTUBE BOILER—SUPERCRITICAL PRESSURE

The monotube design for supercritical pressure is even simpler than the design for subcritical pressure. Since there is no separation of the two phases above 3206 p.s.i., and the increase in specific volume is much less than in the lower pressure range, stability of flow distribution is much easier to maintain. The general arrangement of the tubes is based on the same fundamentals.

The boiler-water separator is omitted, since it cannot function above critical pressure. Continuous blowdown is no longer available and any impurities in the feedwater are either deposited in the boiler or carried over in the steam to the turbine. The need for water of the highest commercially available standard under all conditions of operation is of utmost importance. This can be satisfied with tight condensers, demineralized make-up, and the best possible system for generation, with provision for filtration and demineralization of some portion of the condensate.

In utility-plant development and cycle improvement, rising pressure has gone hand in hand with rising temperature. The limitation of temperature lies in the materials available and their cost and life expectancy, a problem which is now receiving especially concentrated study. The benefit of high temperature may be obtained by increasing the throttle temperature and also by use of either single or multiple-reheat cycle.

Although prediction is difficult, it is likely that

1050°F superheater-outlet steam temperature and double reheat, 1050°F from both stages, will be used for a good commercial design of tomorrow's boiler.

BOILER CONTROLS

The rising cost of labor in the power industry, as in every industry, is one of the factors that encourages the use of high-capacity units and the maximum amount of automatic control. Likewise the trend to high pressure and high temperature calls for the fullest use of automatic control. High-pressure boilers inherently have less storage capacity and are more responsive to changing conditions. High temperature, on the other hand, demands that variations be kept at a minimum to prolong the life of the equipment. The system used for monotube boilers controls feedwater flow, steam temperature, and the essential turbine bypass, providing many of the control features required for safe, efficient, automatic co-ordination of the boiler and turbine.

The basic concept of the boiler-turbine control system is shown schematically in Figure 2, including the turbine-generator set on the right, and the steam-generating unit on the left. The desired end product of the unit is, of course, electrical energy. It is logical then that the generator output be used to regulate the combustion-control system and determine the quantity of fuel and air which is to be supplied to the boiler unit where the energy can be converted and delivered to the turbine in superheated steam. It is the function of the turbine and feedwater-control system to insure that the steam is admitted to the turbine at the proper temperature and pressure to suit the design conditions of the equipment.

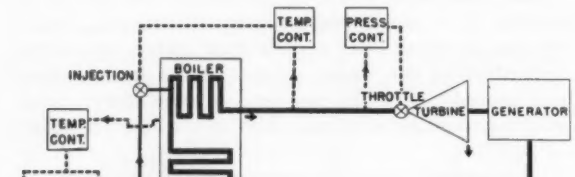

Figure 2. Schematic of basic boiler-turbine control.

Provision has been made, however, in the turbine and combustion controls to transfer to a more conventional control from the direct-acting system illustrated, which is based on European practice and is an innovation to operators in this country.

VARIABLE PRESSURE

Variable or sliding pressure has been the subject of much discussion with the advent of the once-through boiler in the power-generation field and is considered desirable where there is a potential savings if the throttle pressure can be reduced with load. Because of the corresponding reduction of stored energy in the boiler, it is obviously desirable to modify such a system for an adjustable pressure which will retain some of the reserve energy characteristic of constant-pressure operation for quick pickup, while at the same time realizing some of the gain in cycle efficiency incident to reduced-pressure operation at lower loads.

On the other hand, with lower pressure cycles where it is possible to operate one or more turbine valves sequentially, the gain in cycle efficiency with variable-pressure operation all but disappears. In such instances the inherent disadvantages could well offset the advantages.

CONTROLLED CIRCULATION

Based on the requirements outlined, what will tomorrow's boiler look like? Possibly it would be unwise to confine our thoughts to only one, and better to take a look at three typical designs. Certainly controlled circulation has established such an enviable record of performance that it will undoubtedly play a most important part in providing the additional capacity to be installed in the years immediately ahead.

The principal factors which have led to such wide acceptance of controlled circulation are fundamentally related to the limitations of natural circulation in the elevated subcritical-pressure ranges above 2000 p.s.i. at the turbine throttle, and particularly as these exist in very large units.

Boiler structures are now becoming of such large physical size, at 200,000 to 600,000 kw, that controlled circulation (or once-through circulation) can be of considerable advantage in establishing positive flow through all furnace and boiler tubes prior to firing the unit. Expansion movements of as much as 8 in. vertically and 5 in. horizontally obviously become a matter of significant consideration. Controlled recirculation also makes possible the uniform heating and cooling of the thick-walled heavy drums in these large high-pressure units which are so important in both normal and emergency starting and shutdown.

Considerable operating service with high-capacity recirculation pumps of both injection and submerged-motor types has now eliminated the concern first felt on the use of this equipment.

The use of small-diameter relatively thin wall tubes is a natural development in the use of forced circulation. About 40-p.s.i. pressure head is provided in the controlled-circulation pumps, or almost 5 to 7 times that normally calculated as available to produce the circulation in a large natural-circulation boiler at 2400-p.s.i. turbine-throttle operation.

TYPICAL MONOTUBE INSTALLATION

Typical of the possibilities of the monotube once-through installation at high subcritical pressures is

the Portland unit of Metropolitan Edison Company. This unit has a capability of 175,000 kw with a throttle pressure of 2520 p.s.i. with moderate steam and reheat temperatures of 1050°F.

The economic analysis of the supercritical cycle gives reasonably clear indication that within the next few years a large proportion of new utility units will be for operation in the range of 3500 p.s.i.

The 358-mw No. 2 unit for Eddystone was designed for this cycle. The boiler will generate 2,178,000 lb. of steam per hour with 3500 p.s.i. and 1050°F at the turbine throttle. There are two reheats, the first at about 1000 p.s.i. and 1050°F, the second at about 300 p.s.i. and 1050°F.

II—Modern Steam Generator Designs

THE FIRST commercial supercritical-pressure steam generator placed in service at the Philo Plant of Ohio Power Company on March 20, 1957, has surpassed all expectations for boiler availability, performance, ease of operation, and freedom from troubles pertaining to the once-through principle. The possibilities for reduced capital and operating costs, and the simplicity of operation with excellent steam-temperature characteristics over the operating range point to an increasing trend toward subcritical and supercritical units of this type in the near future.[1, 2]

STRUCTURAL-DESIGN TRENDS

The trend toward the general use of single-boiler single-turbine installations in central stations has resulted, in part, from the fact that boiler availability increased to a level comparable to turbine availability. This has been remarkably improved by the use of tangent-tube furnace-wall and flat-stud tube-wall assemblies, eliminating almost completely the use of refractory materials which are subject to excessive maintenance during outages.[3]

In a typical modern furnace-wall section of tangent-tube construction the basic sealing requirement is provided by a gas-tight seal-welded inner casing in contact with or in close proximity to the pressure parts. A single layer of block insulation provides an optimum solution to the temperature-reduction problem. With this combination, the outer covering no longer acts as a seal, and light-gage metal-lagging designs have been developed and are coming into use which are suitable for both indoor and outdoor installations.

Complete elimination of rolled-tube joints in drums and headers of high-pressure units, almost complete elimination of gasketed joints, and improved welding techniques in both shop and field are other major factors contributing to high boiler availability and reduced maintenance costs.

PULVERIZED-COAL-FIRED DESIGNS

Conventional-fuel-burning steam-generating plants patterned after present-day progressive designs will provide, by far, the major portion of the predicted capacity increase during the next 15 to 20 years. A modern steam generator which has been in service at the River Rouge Plant of the Detroit Edison Company since February, 1956, delivers 1,720,000 lb. of steam per hour to a 260,000-kw turbine, is one of the largest in service to date, and typifies the trend to higher duty and higher capacity. Operating at 2051 p.s.i., the primary steam temperature, 1050°F, and the reheat steam temperature, 1000°F, are controlled over a wide load range by means of gas recirculation in conjunction with gas proportioning over the superheater and reheater surface, and superheater spray attemperation. This pulverized-coal-fired boiler features a single furnace with a division wall and a dry-bottom cross-hopper having a shielded arrangement. Spaced circular-type burners are arranged to fire coal, oil, or blast-furnace gas in conjunction with coal.

The slag-tap pulverized-coal-fired unit, installed in the Burlington Station of the Public Service Electric and Gas Company, was placed in commercial service in October, 1955. Designed for 2700 p.s.i., this unit delivers 1,225,000 lb. of steam per hour to a 180,000-kw turbine at 2390 p.s.i. and 1100°F at the superheater outlet, with reheat to 1050°F.

Wide-spread radiant-superheater platens provide a much flatter steam-temperture characteristic than could be attained with only convection surface. The high primary and reheat steam-temperature conditions require a very high percentage of the total-unit heat absorption in the superheater and reheater; and gas circulation to temper the combustion gases and prevent slagging of the radiant platens is a design feature which has proved satisfactory in operation. Without gas tempering, an appreciable load reduction is required on the unit to prevent excessive slagging of the radiant-superheater platens.

The foregoing units are fired by circular-type burners. Capable of firing pulverized coal, oil, and gas, singly or in combination, the turbulent mixing characteristics of this type burner provide excellent ignition stability over a wide load range on all fuels. The burner throats are completely water-cooled, being covered only with a thin layer of refractory, retained by pin studs, to provide smooth throat contours so that maintenance requirements are practically eliminated. Ring-type gas burners which are inaccessible for repair or cleaning while in service, have been replaced by new multispud gas burners which are removable for either cleaning or repair with the unit in service. Automatic oil and gas-lighter development work has resulted in designs

insuring dependability; and all burner adjustments, during start-up, shutdown, and normal operating periods can be remotely regulated from centralized control rooms.

CYCLONE-FURNACE DESIGNS

The cyclone furnace is the result of continued efforts to improve the methods of burning fuels, to better the functional design and performance characteristics of boilers, and to minimize the problems of air pollution.[4] The cyclone furnace is a horizontal water-cooled cylinder of tangent-tube construction in which the complete combustion of fuel takes place at inputs per cyclone ranging to 500,000,000 B.t.u. per hour. Crushed coal, gravity fed to the burner, is mixed with the primary air stream and deposited on the slag lining where rapid surface burning by a high-speed tangentially admitted secondary air stream occurs. A new development, the vortex coal feed, greatly reduces the back pressure on the coal feeders and eliminates costly rotary seals. Depending on the primary-furnace design, 80 to 85 per cent of the ash content of the coal is tapped from the cyclone and then through primary-furnace floor openings in molten form to water-filled slag tanks for quenching and disposal. Refiring of fly ash, collected from the products of combustion after discharge through the cyclone re-entrant throat and passage through the boiler unit, is utilized to completely eliminate the nuisance of fly-ash handling and disposal.

The cyclone method of firing simplifies the burning of gas and oil at ratings and with performance equal to coal firing. Gas is fired through ports integral with the secondary-air entrance and oil is introduced at low pressure, 50 p.s.i., by a roof-type burner which consists of a perforated pipe across the secondary air ports and which remains in firing position at all times. During an interruption in coal supply, the roof-type burner can be placed in full-load service in a few seconds. A gun-type oil burner, inserted axially through the burner front, is used for long-term oil-firing periods but must be removed during coal-firing periods to prevent tip fouling. The burning characteristics produce a longer flame travel from the re-entrant throat which tends to increase steam-temperature control range.

This significant advance in the art of burning coal numbers among its advantages the ability to handle a wide variety of coals with very low excess-air requirements and low carbon loss; appreciably reduced stack-gas dust loading; thus facilitating the reduction or elimination of dust collectors; and the elimination of fly-ash erosion of tubes. The cyclone method of firing reduces appreciably the number of interlocks, controls, and lighters since a single cyclone can burn about four times the amount of fuel fired by typical pulverized-coal burners. Simplicity and reliability of operation are excellent and, to date, a furnace puff or explosion has never occurred on a cyclone-fired unit when burning coal.

This principle of firing is now being applied to a high-capacity steam generator being erected at the Joliet Station of the Commonwealth Edison System. Designed to generate 2,200,000 lb. of steam per hour at 2100 p.s.i. and 1055°F, with reheat to 1005°F, this unit will serve a 325,000-kw turbine. Nine cyclone furnaces are arranged for opposed firing in a single furnace and this opposed-firing feature will be utilized frequently as unit sizes increase. The arrangement of the unit features completely drainable pressure parts, and the trend toward increased use of wide-spaced radiant-superheater and reheater platens is incorporated in the design to provide relatively constant steam temperature over a fairly wide load range.

PRESSURE FURNACES

Pressure-furnace operation of steam generators in central stations continues to gain in popularity. This significant design advance was made possible by the development of a gas-tight inner casing (now a feature of most modern nonpressure units as well). The success of this feature was contingent on the casing temperature closely paralleling the pressure-part temperatures under all operating conditions so as to reduce stress. Compared to the older conventional settings with their attendant room-air infiltration, a pressure-cased boiler operating with only forced-draft fans offers a gain in boiler efficiency in the order of 0.3 to 0.5 per cent. Pressure-furnace operation offers definite advantages such as the reduction in total fan power, due to the smaller volume of relatively cold air handled by the forced-draft fan, and the elimination of induced-draft fans and their attendant maintenance. Operation is simplified since it is not necessary to maintain a balance between the forced and the induced-draft fans. A definite trend to pressure-furnace operation and elimination of induced-draft fans prevails on gas and oil-fired units and on units with cyclone furnaces since they are ideally suited for this mode of operation. This trend is not too apparent in pulverized-coal-fired installations due, in part, to the necessity of double valving a large number of burner lines and other sealing requirements.

STEAM PURITY

With the trend to higher and higher operating pressures, the problem of turbine-blade fouling by tenacious silica deposits becomes acute, resulting in loss of efficiency and capacity. Mechanical separating equipment is ineffective since silica is carried out of the boiler in gaseous solution. Silica solubility in steam increases rapidly with higher saturation temperatures—an increase in operating pressure from 2000 p.s.i. to 2800 p.s.i., for example, increases the vaporous silica in the steam five times.

Generally, 0.025-ppm, or less, silica content in steam is considered good practice. Drastic reduction in boiler-water silica concentration at high pressures would be required to comply with this limita-

tion. To decrease blowdown requirements and the need for almost silica-free water, units operating at high pressures are equipped with highly effective steam washers. Cyclone steam separators provide for the primary separation of steam from the steam-water mixture, and primary-steam scrubbers above the cyclones remove entrained-moisture droplets. The steam then enters the washer arrangement passing consecutively through a perforated steam-distribution plate, a stainless-steel wire pack, a perforated water-distribution plate, and a corrugated secondary-moisture eliminator. Wash water is distributed above the upper distribution plate and flows downward countercurrent to the steam flow.

The extensive surface of the wire pack, providing very large surface contact between steam and wash water, and violent agitation of wash water on the trays by the high steam mass flow results in maximum silica reduction over a wide load range. At full boiler load, with the wash-water flow approximately 5 per cent of the feedwater flow, the steam washer will remove 75 per cent of the steam-dissolved silica. Water for the steam washer is taken from the feedwater header and the control system is simple and requires no operator attention.

UNIVERSAL-PRESSURE BOILER DESIGNS

Thermal gains in steam-electric plants can best be effected by increased steam pressure and temperature. One of the greatest advances in this regard was the development of the first supercritical-pressure boiler for power generation in this country. The cyclone furnace unit at the Philo Plant serves a 125,000-kw turbine and generates 675,000 lb. of steam per hour at a pressure of 4500 p.s.i., and a temperature of 1150°F. The steam is reheated twice—to 1050°F and 1000°F.

Two considerably larger supercritical-pressure units are scheduled for installation in the American Electric Power System. Each unit will serve a 450-mw turbine and deliver 2,900,000 lb. of steam per hour at 3500 p.s.i. and 1050°F, with two stages of reheat to 1050°F. The Sporn Plant unit is designed for pulverized coal firing, and the Breed Plant unit of the Indiana-Michigan Electric Company is designed for Cyclone-Furnace firing.

In the subcritical-pressure range from 2400 to 3000 p.s.i., in unit sizes of 175,000 kw and larger, the once-through unit has definite advantages, as compared to drum-type boilers, due to elimination of the heavy drums and downcomers, the reduction in support steel, the over-all weight, and the simpler controls.

Two Universal-Pressure steam generators scheduled for installation at the Huntington Beach Station of the Southern California Edison Company will each supply 1,638,000 lb. of steam per hour to a 200,000-kw turbine at 2450 p.s.i. and 1050°F with reheat to 1000°F. Firing is by the circular-type oil and gas burners, arranged in two levels in all four furnace walls.

III—Trends in Present Day Boiler Design

ONCE-THROUGH steam generators should become more numerous in the U. S., since they have advantages in lower weight, control of steam temperature over full load range, possibility of variable pressure and temperature operation, and rapid start-up and shutdown procedures. On the other hand, their selection over the more conventional types of boilers involves economic consideration of requirements for more sensitive control equipment, greater pump power, high-purity feedwater, more condensate storage, and the necessity for a bypass line heat-exchanger system.

Operating experience with supercritical pressures that will affect future designs of once-through steam generators is being obtained in the Foster Wheeler Cartaret pilot plant.[5] This steam generator has a capacity of 2000 lb. per hour of steam operating up to 5500 p.s.i. and 1200°F. Since 1956 this plant has operated 5000 hours. Feedwater treatment and corrosion rates have been investigated, and control systems for commercial once-through steam generators have been studied.

PRINCIPAL TYPES

Once-through steam generators for supercritical as well as subcritical pressures are of two principal types, the Benson and Sulzer. A Benson-type design for subcritical pressure has a capacity of 1,800,000 lb. per hour at 2400 p.s.i. and 1050°/1050°F. Important in correct design of this type unit are: (*a*) Provision for frequent collecting, mixing, and redistribution of fluid along the complete flow path from inlet to outlet; (*b*) low furnace heat-absorption rates and uniform heat distribution to fluid circuits; (*c*) final evaporation or conversion zone located in low gas-temperature sections; (*d*) drainable pressure parts.

At supercritical pressures, and at subcritical pressures within 400 lb. of the critical pressure of 3206 p.s.i., forced circulation is required for the once-through type of unit. At lower pressures a choice can be made between natural circulation and pump circulation. Fortunately, it has been the practice for many years to calculate the circulation of new boiler designs. This background and the close agreement that has been obtained between predicted circulation and actual measurement permits the designer to confidently predict and design for natural circulation at high subcritical pressures with ample safety margins in spite of the reduced circulating force available.

Criteria used as basis for natural-circulation design are: (*a*) A limitation on the per cent steam by volume in the steam-water mixture leaving heated portions of furnace or boiler tubes at maximum capacity, usually 65 to 85 per cent steam by volume depending on pressure and type of circuit; (*b*) velocity of water entering the heated portion of tubes at least 1 to 2 fps, also depending on pressure and type of circuit; (*c*) separation of steam and water in drum to give steam-free water in downcomer supply; (*d*) segregation of circuits with differing absorption rates.

An advantage often cited for natural circulation is its self-compensating effect. If absorption rate is somewhat higher than predicted because of miscalculation of friction loss, localized hot spots, or other unforeseen circumstance, the average density of the circuit is reduced and flow to the circuit increases.

As operating pressures increase there is a rapid increase in the per cent of silica that will vaporize and be carried over with the steam to deposit on turbine blades. Generally acceptable limits of silica in steam are below 0.025 ppm, achieved through close control of silica concentration in boiler water by feedwater treatment and blowdown. Mechanical separators will not remove silica after it has vaporized in a boiler, but it can be reduced by washing with incoming feedwater. Being low in silica it will dissolve a portion of silica vapor from steam passing through it, depending on degree of mixing and the purity of feedwater. The dual-circulation principle is another approach,[6] whereby concentration of silica (and total solids) in the boiler water from which steam is generated can be maintained at much lower values. In a conventional boiler, all boiler water comes from a common downcomer system and results in uniform solids concentration entering all sections of the boiler. In a dual-circulation boiler, the arrangement of feedwater-intake and downcomer-supply circuits is such that the steam generator has two separate circulating systems, and solids concentrations are no longer uniform. The boiler-water concentration in the secondary section is maintained at or above the recommended limits of the American Boiler & Affiliated Industries. Due to the dual-circulation principle, the resulting boiler-water concentration in the primary will be about 50 to 25 per cent less depending on the design. Steam generated from the low-concentration primary section will have correspondingly less silica vapor and total solids. Steam generated from the higher-concentration secondary section will be all or partly condensed by contact with incoming feedwater so that this steam contributes little if any to the impurity of saturated steam passing out of the boiler.

An 1850-p.s.i. dual-circulation reheat unit has two primary sections, the first being furnace side and division walls, and the second being furnace rear wall. The secondary section is comprised of boiler and wall surface in the convention pass. Connections for recirculation between the sections permit flexibility in chemical control of various constituents. Test results show less than 0.005 ppm silica in steam while silica range in primary was 0.2-3 ppm and in secondary 5-10 ppm.

TEMPERATURE

The trend toward increasingly higher steam temperature is at present temporarily halted by lack of suitable materials. Metals for steam temperatures of 1050°F are generally accepted as giving satisfactory service life, although temperatures higher than this are being employed.

The development of steam generators which provide wide-range and close control of steam temperatures is continuing. It is possible to design large steam generators which provide steam at any desired temperature level between maximum design temperature and saturation, or reheat inlet temperature, over wide ranges of load. This involves a multiple-furnace design having separate superheat, reheat, and water-evaporating furnaces, but cost is not competitive.

There are a number of principles which can be applied to more conventional steam generators in order to provide wide ranges for constant temperature control. These include use of radiant surfaces for superheating and reheating, gas recirculation, and flame-shift methods. Radiant steam-heating surface, in use by the author's company for over 40 years, presents the steam generator designer with two useful tools.[7] The steam temperature leaving this surface decreases with increasing load. Also, by placing a portion of the superheater or reheater surface in the furnace, the steam-heating duty remaining for convection surface is less and gas temperature leaving the furnace can be reduced below that for all-convection surface.

In a typical design, the radiant superheater forms the front firing wall, hopper slope, and roof. Other recent designs have incorporated radiant steam-heating surface in full and partial division walls, in side-wall panels, or interspersed with water-wall tubes in side walls.

CAPACITY

There is a continuing trend to ever-increasing capacity in a single unit. Some of the larger units are of double-furnace type while others retain the single-furnace form. In either case, design for the larger capacities requires special attention to support of the long spans, drum size, and lengths, accessibility for cleaning, and uniform distribution of both heat and flows throughout the unit.

Size of such auxiliaries as pulverizers, firing equipment, regenerative air heaters, and fans are also increasing. Designs for ball-mill pulverizers, for example, are being prepared with capacity approaching 100 ton per hour on a favorable coal, although the largest mill being built has a capacity of 46.5 tons per hour.

Large capacity has increased the incentive of designers to proportion heating surfaces for optimum use of available temperature levels. As the total amount of surface increases, substantial dollar savings can be realized by placing heating surface in the most advantageous temperature zone at the expense of some complication.

A recent example is a 2,350,000 lb. per hour reheat steam generator with 2200 p.s.i. 1010°/1010°F conditions. Saturated-steam connections from the drum form the rear wall of convection pass and enter a section of convection superheater; this places the lowest steam temperature in the lowest gas-temperature zone preceding the economizer, and results in optimum-mean-temperature difference. Steam then passes through the radiant superheater located on front wall and roof of the furnace. This incorporates the radiant characteristic into the superheater, reduces the gas temperature required at furnace exit, and reduces the duty required of the finishing section of the superheater. As a result, the gas temperature entering the reheater is at high enough level to give advantageous heat-transfer rates without high furnace-exit temperature. The amount of superheater surface in the first convection section, radiant section, and the finishing section can be apportioned for optimum balance of metal temperature and mean temperature difference between gas and steam.

With this arrangement, control of steam temperatures is obtained by proportioning gas flow between the two parallel gas passes by means of dampers located at the outlet of the economizer, plus a feedwater tempering spray or condenser in the superheater. The duty represented by the difference between the uncontrolled temperature and the controlled temperature is shifted from superheater by control of gas flow in the two parallel passes. By this means the control range of the reheater is increased without additional surface.

EFFICIENCY

The various means available or contemplated for attaining higher steam-generator efficiency through lower exit-gas temperature usually involve adding heat-recovery surface in the temperature range where corrosion and plugging of the surface will occur. Evaluation of the initial cost and operating expense of such surface has so far justified the resulting higher efficiency in only a few power stations.

Provisions must be made to maintain effectiveness of this heat recovery without removing the steam generator from service. This usually requires a system for inservice washing, and fabrication of exposed metal surfaces from alloy steel, cast iron, or noncorrosive material clad to metal surface. Investigation now being pursued, that appears to provide greatest benefits in this regard, is ammonia injection into the flue gases to reduce corrosion by preventing the formation of H_2SO_4 from sulfur-bearing fuels.

ERECTION

In recent years there has been an increased use of shop prefabrication in order to realize the savings that result from lower shop-labor costs compared to field-labor costs, from the superior quality control obtainable in a manufacturing plant, and time saved on field erection. Greater use is being made of panelized tube-wall sections, sectionalized superheaters, reheaters and economizers and flue and duct assemblies, and other prefabrication techniques.

A recent development in this direction is the integral-fin wall construction shown in Figure 3 with the fins welded continuously between open-spaced tubes to form a solid wall. An integral-fin section using ¾-in. wide fins has only about 80 per cent of the weight of a comparable panel of tangent tubes with skin casing.

Two units with the integral-fin wall have been placed in operation this year. Each has a capacity of 445,000 lb. of steam per hour at 1360 p.s.i. and 950°F.

Three natural-gas-fired pressurized reheat units which also make use of the integral-fin design are now being fabricated. These units range in capacity from 1,550,000 to 2,150,000 lb. of steam per hour.

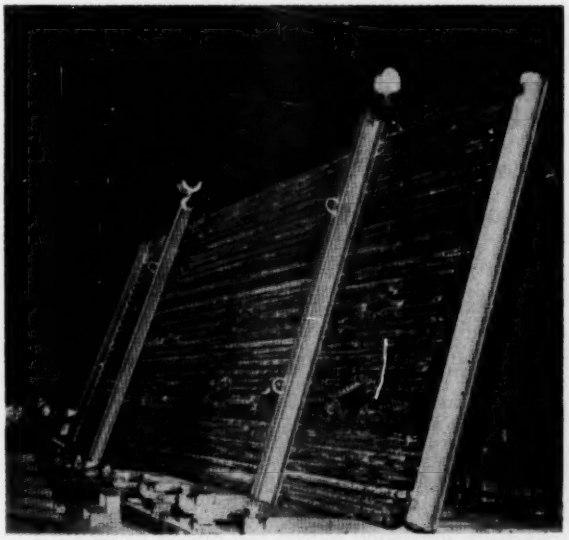

Figure 3. Factory-assembled water-wall panel of the integral-fin type ready for shipment to the field.

COMBINED GAS-STEAM CYCLES

The use of gas turbines in power generation has received considerable study in recent years, including their use in combination with steam boilers. There are at present few installations of this kind, but we believe their use will find increased favor in

years to come. The steam generators used in the combined gas-steam cycle present interesting design considerations.

One such cycle makes use of exhaust from gas turbines as combustion air supply for a conventional steam boiler.[9] The boiler can be designed for the firing of solid, liquid, or gaseous fuels. A steam generator for this cycle is now nearing completion for a southern utility as part of a 40,000-kw combined gas-steam cycle.

Another type of steam generator being considered for gas-turbine applications is the supercharger boiler,[10] which takes the place of the gas combustor. Atmospheric air compressed to over 70 p.s.i. enters the supercharged boiler where combustion of fuel takes place. Gas discharge from the boiler is at the high pressure and temperature required to operate the gas turbine.

Supercharged boilers in operation at present have capacities on the order of 150,000 lb. of steam per hour and are limited to use of natural gas or light-oil fuels. Some progress has been made in overcoming the problems associated with firing residual oil and coal, but these fuels are not yet ready for commercial use in supercharged boilers.

Figure 4 shows a reheat unit proposed for 133,500-kw installation and would produce 735,000 lb. of steam per hour at 1800 p.s.i., 1000°/1000°F. This is a natural-circulation, twin-furnace boiler using differential firing to control primary and reheat-steam temperature. The supercharged boiler is characterized by furnace-heat-release rates and heat-absorption rates on the order of 3 to 4 times higher than normal for conventional steam generators, and therefore occupies considerably less space per lb. of steam generated than a conventional boiler. This unit, for example, consists of two 15-ft.-ID shells and has a floor space including supporting steel and platforms of 30 × 50 ft. with an over-all height of 100 ft.

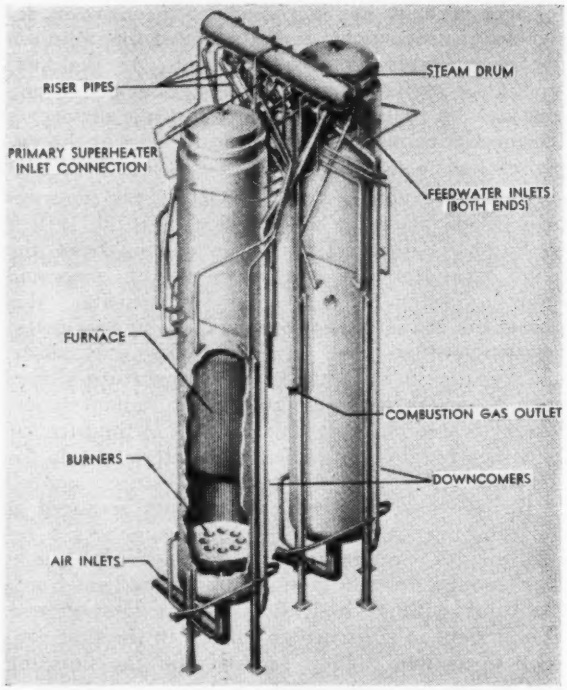

Figure 4. A proposed design of the supercharged type for the combined gas-turbine steam-turbine.

IV—Design Requirements for Steam Generation with High Cost Fuels

THE DESIGN of steam-generating units for northern areas of high fuel cost involves a number of existing problems. Among these are: The necessity for design for more than one fuel; the economic justification of high turbine-cycle efficiencies involving the use of higher pressure and temperatures; the need for greater efficiency of the steam-generating equipment; the necessity for continuous availability; the requirement that the units operate at the maximum efficiency within the design capacity.

MULTIFUEL DESIGN

Areas of high fuel cost usually have more than one fuel which can be economically justified, although in the past this has involved considerable additional investment. Designs are now available which minimize this additional investment and provide for the efficient use of a variety of fuels with no expensive alterations when changing from one fuel to another. In these, the same furnace performance is available with different fuels and consequently the same range of steam-temperature control is possible without costly auxiliary equipment.

The designer's problems when designing for two or more fuels begin with the furnace, continue with the superheater and reheater and on through the unit to the airheater gas discharge.

FURNACE AND SUPERHEATER DESIGN

The furnace requires sufficient cooling to reduce the furnace exit gases well below the ash-softening temperature. The increase in steam temperatures to 1000°F, 1050°F, and even higher, focuses attention on the variation in tube-metal temperatures across the width of a furnace. If the metal is to have a reasonably long life this variation must be kept to an absolute minimum, and the methods of firing must produce a uniform gas distribution. The problem is more pronounced as the capacity of the units

increase and the furnaces become wider, requiring close attention to the maximum furnace-exit temperature as related to the average.

One satisfactory solution is the use of water-cooled platens within the furnace section. These platens are formed by rows of tangent tubes spaced across the upper furnace to form a number of passages which can be variously disposed to suit the design requirements. The location is above the zone of active burning so there is no interference with combustion. This construction furnishes the designer with a flexible tool in designing for a required furnace exit-gas temperature by placing more or less radiant surface between the active combustion zone and the superheater entrance. The platens also absorb heat at intervals across the furnace between the walls, thus reducing the amount of gas-temperature variations and the possibility of slagging. Superheater tube-metal temperature measurements with this type of construction range from 1035° to 1050°F, with 1000°F final temperatures even in wide furnaces.

Furnace volume is important in providing sufficient space for flame development and completion of combustion. It is also important in providing the necessary time for the water-cooled surface to cool the gases before the furnace exit. If the furnace-residence time is too short, the gases cannot cool to the required level. The designer when confronted with two or more fuels must design a furnace for the one requiring the greater volume. The larger units in the New England area burning pulverized coal and oil should have furnaces designed for a volume heat-liberation rate of approximately 18,000 B.t.u. per hour per cu. ft. of volume and no higher than 85,000 B.t.u. per hour per sq. ft. of projected furnace surface.

The maintenance of steam and reheat temperatures over an economical range when designing for two fuels requires care in selection of the amount and disposition of superheating surface. The amount of required surface varies with the different fuels and is usually least for natural gas and highest for oil with coal falling between. It might be that no economy would result from operating with a second cheaper fuel, such as oil, if the superheater surface has been sized for coal, because the resulting lower steam temperatures would tend to offset any saving. It is essential to design for approximately the same temperature range with all available fuels. This problem is simplified by the use of intermediate radiant surface in the form of a platen superheater in the furnace. The superheater surface is disposed as radiant platens widely spaced across the width and water-cooled platens are used in the upper furnace.

HEAT RECOVERY

The design of heat-recovery equipment, particularly the air heaters, requires special care where two fuels are available. The high-cost fuel makes low air-heater exit temperatures mandatory, but the heater surface must be protected against corrosion, especially that resulting from attack by sulfur compounds. If oil with sulfur as high as 3 per cent is burned, larger amounts of air recirculation requiring larger fans, larger heating coils, or possibly a combination of both will be required.

Careful selection of economical operating pressures and temperatures for the turbine and steam-generating equipment is always essential but even more important when considering units subject to high fuel cost. The general trend to higher pressures and temperatures is most marked where fuel prices are relatively high.

Similarly, the need for continually higher steam-generating efficiency is obvious and mandatory in high-fuel-cost areas. As we become intrigued by the thermal gains obtainable with high pressures and temperatures, we tend to forget the less spectacular but equally valuable gains which can be obtained by reducing the final exit-gas temperatures.

AIR-HEATER PROTECTION

A decrease in air-heater exit-gas temperature is accompanied by an increase in the rates of heating-element corrosion and deposit formation requiring improved cleaning techniques or corrosion control if more efficient air-heater operation at lower exit-gas temperatures is to result. In connection with regenerative air heaters, there has been constant improvement in the technique of cleaning by using air or superheated steam as the blowing medium and by the extensive use of water for washing the heating elements. Techniques and equipment are available for in-service washing which insure maintenance of air-heater efficiency at low exit-gas temperatures.

A somewhat different approach is used in Unit No. 3 of Detroit Edison Company's River Rouge Plant.[11] This installation is equipped with four Ljungstrom air heaters in a series arrangement which are furnished with enameled cold-end heating elements for protection against corrosion from the exposure to sulfur compounds and moisture at temperatures as low as 200°F.

The installation of a stack economizer in the cold gas from the air heater as part of a closed system circulating hot water through air-heating coils at the entrance to the main air heaters is another solution to the problem of operating continuously at low exit temperatures. Such a system acts to raise the average metal temperature of the cold-end heater elements and to reduce the corrosion rate. This water can be at a pressure rate which will permit the use of a cast-iron economizer to minimize the corrosion.

Further development in the search for materials to resist this low-level corrosion is proceeding throughout the industry. The use of glass or ceramic materials appears to hold some promise.

AUXILIARY POWER

The over-all station efficiency is affected by a number of relatively small factors, important in the aggregate.

The magnitude of the auxiliary power required to move the air and gas through modern units suggests the possibility of substantial savings. Operation of the system as a full-pressure design without induced-draft fans will reduce this power appreciably when compared to balanced-draft operation for two basic reasons: First, the volume is less because only ambient air is handled; second, airfoil blade fans with efficiencies ranging from 84 to 88 per cent can be used. Savings with pressurized operation may run as high as 30 per cent at the operating load as shown in Table I. This shows a comparison of fan-power and motor requirements for a unit having a capacity of 1,200,000 lb. of steam per hour. Savings in installed horsepower and auxiliary equipment as well as initial investment and induced-draft-fan maintenance costs are obvious.

The continuing trends to greater capacity and pressures in steam-electric generating stations have created a new interest and need for high-speed steam-turbine drives for boiler-feed-pump service to decrease the demand for auxiliary power.

Similarly, the use of turbine-driven forced-draft fans on pressure-fired units using steam from the low-temperature-reheat steam line would appear economical. Certain applications might require small auxiliary drives for starting from a cold condition.

TABLE I— *Comparison of Full-Pressure Operation With Suction Design*

	Suction operation		Pressure operation	
Steam, lb. per hr.	1,200,000	Fan design	1,200,000	Fan design
Air, lb. per hr.	1,574,000	1,810,000	1,574,000	1,810,000
Air temperature, °F	80	80	80	80
Air volume, cfm	366,000	420,000	366,000	420,000
Total static pressure, in. water gage	7.9	10.4	17.2	22.5
Input to fans, hp	650	890	1350	1770
Motors	Two, 450 hp		Two, 900 hp	
Induced-draft fan data				
Gas, lb. per hr.	1,692,000	1,946,000		
Flue-gas temperature, °F	295	315		
Gas volume, cfm	528,000	622,000		
Total static suction, in. water gage	9.4	12.4		
Input to fan, hp	1310	1691		
Motors	Two, 900 hp			
Total input to fans, hp	1960		1350	
Horsepower saving			610	(31%)
Motor saving			Two, 450 hp	

AVAILABILITY AND EFFICIENCY

The higher justifiable efficiencies and the resulting increased investment costs stress the need for maximum availability. There is no gain obtainable from more efficient equipment if it remains idle. A furnace suitable for the lowest-grade-available fuel and adequate protection of the heat-recovery equipment are essential for economy. Other factors affecting availability are: adequate circulation, careful selection of superheater and reheater metals, wide spacing of heating surfaces to maintain free gas passage, and suitable characteristics of pulverized-coal equipment.

Again there is little justification for highly efficient design unless it is possible to operate the units continuously at their maximum design efficiency.

AUTOMATIC SYSTEMS

The manufacturers of controls and instruments are continually working toward automatic systems and that other long-cherished ambition of every designer—an automatic push-button plant. Interesting steps toward this achievement are the automatic-data-collection systems now in operation at the Sterlington Station of Louisiana Power and Light Company and Neches Station of Gulf States Utilities.

The following information, published [12] on the equipment at the former station, indicates the trend.

The ADC (more popularly known as DADIT) is installed on 200-mw and 43-mw rated units. It is used to:

(1) Scan 250 temperatures during operation of the units at a rate of 5 points per sec. The points scanned are: the bearings and bearing oil on the main unit and auxiliary equipment; boiler, superheater and reheater tubes; condensate steam; transformers; and other points important to reliable plant operation. Should any of these temperatures exceed a preset alarm point, the digital reading will be printed out and an annunciator will sound an alarm and identify the off-normal point. Printout of

the off-normal will continue during each scan cycle until the temperature is brought back to normal or the equipment in trouble is taken out.

(2) Log out in digital form any of the scanned temperatures on demand from the operator.

(3) Periodically log, on prepared log sheet, another 100 plant readings necessary for accounting and trouble-shooting records. These include flow, pressure, temperature, and heat rate; electric amps, volts, and mw.

The DADIT system consists of a computer with a 1024-word memory. This equipment can be programmed to give any set of readings required, and can be used to obtain corrective control action for various off-normal points. This latter feature will be utilized on this unit only for testing and experimenting with the computer. The computer consists of solid-state circuitry with some mercury relays.

The most important reason given for installing this unit is to provide operating experience with a type of equipment that will be used in the completely automatic power plant, namely, the continuous computer scanner. The eventual goal is steam and electrical generation operated by automatic equipment and supervised and controlled by the computer which will take over sequential start-up, operation, and shutdown. All maloperation which might normally occur will be avoided and such a system produce test efficiencies as a daily routine.

CONCLUSION

No spectacular scientific breakthroughs have been described, but rather individual design features of special significance for plants in high-cost-fuel areas. Progress in steam generation and combustion of fuel will be made in the future, as in the past, by continuous improvement and refinement of numerous details.

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Ultra-sonic energy, or inaudible sound waves, are finding application in a wide variety of uses. Industrial drilling and grinding techniques have been developed. Chemical reaction rates have been influenced and techniques for emulsifying and homogenizing materials have been developed. The use of a suitable transducer in a solvent tank has indicated that remarkable results can be obtained in cleaning and de-greasing, suggesting the possibility of wide industrial application, since cleaning operations are costly and time-consuming in many processes.

—from SCIENCE NEWS LETTER
January, 1959

The first of an order of eleven passenger ships for the Soviet Union has been placed in service. The ship was built in East Germany at Wiemar. It is propelled by twin, six-cylinder Diesel engines developing about 3950 hp each. Designed to carry 340 passengers at a service speed of 18 knots, the vessel and its sisters are expected to be used principally in the Baltic and North Sea areas. The ships will have the following principal features:

Length, overall	400 ft. 9 in.
Beam	52 ft. 6 in.
Draft	16 ft. 11 in.
Gross tonnage	4722
Displacement, tons	5443
Speed, knots	18
Endurance, miles	4200

This order, coupled with negotiations underway for construction of six 13,000 ton passenger ships, is indicative of Russia's growing potential as a maritime power.

—from TRANSACTIONS OF THE INSTITUTE OF MARINE ENGINEERS
January, 1959

In a paper read at the Second Atomic Energy Conference at Geneva, hitherto unpublished data on the Soviet nuclear-powered icebreaker was revealed. The ship's dimensions are:

Length, overall	439 ft. 6 in.
Beam	90 ft. 6 in.
Draft	30 ft. 2 in.
Speed	18 knots

Three pressurized water systems are installed; any two systems are sufficient to produce full power. Each pressure vessel is about 6½ ft. in diameter and 16½ ft. high. Two coolant loops are associated with each vessel, operating at a maximum pressure of 2840 psi and a combined flow rate of 590 cfm. Throttle steam conditions are 410 psig and 590°F. Four main turbines drive generators to produce a total of about 16000 KW at full power. Triple screws are installed, the center screw being driven by a 19,600 shp propulsion motor and the side screws each being driven by a 9800 shp motor.

—from THE JOURNAL OF BRITISH SHIPBUILDING RESEARCH ASSOCIATION, January, 1959

CAPTAIN E. A. WRIGHT, USN

THE BUREAU OF SHIPS: A STUDY IN ORGANIZATION

THE AUTHOR

is an Engineering Duty Officer who has served in the Bureau of Ships and several of its field activities, and in the U.S. Atlantic and Pacific Fleets.

EDITOR'S NOTE: *Proceeds from this paper were contributed to the Navy Relief Society.*

AUTHOR'S NOTE: *Part I of this paper, which appeared in the February 1959 issue of the* JOURNAL, *examined the evolution of organization in the Navy Department for researching, designing, building and maintaining ships of the U. S. Navy, with particular emphasis on the reorganization of the Bureau of Ships in 1958. Part II, which follows, discusses organizational principles and the alignment of the present Bureau of Ships organization with some of these principles.*

PART II ORGANIZATIONAL PRINCIPLES

ORGANIZATION means many things. Organization means the relationships between functions, physical factors, and personnel for the accomplishment of a desired objective with maximum economy and effectiveness (23). It is a two-way street, the dividing downward of responsibilities, authorities, and resources, and the synthesis upwards of coordination, accountability, and results (5). Organization deals with individuals and their relations; it is the sum of individual endeavors (24). Principles of organization are the distillation of experience in generalized form. Principles are guides, but only guides. Principles are beautiful tools but, like the surgeon's scalpel or the dentist's turbine drill, they should be used only in thoroughly discriminating hands.

The Bureau of Ships organization will be lightly overlaid on a discussion of apparently-applicable basic principles. These principles have been garnered from many sources, screened for applicability, discussed briefly in the following paragraphs, and melded in conclusions at the end. Especially helpful has been the authoritative work of Gulick (1), Seckler-Hudson (5), Livingston (23), Brown (24), Terry (25), Beishline (26), Anthony (27), Drucker (28), Mooney (29), Mayo (30), and Barnard (31). The purpose of the analysis is to derive those fundamentals seemingly essential to the present and future organization of the Bureau of Ships, or to any future organizational segment with the mission SHIPS.

THE PURPOSES OF ORGANIZATION

The purpose of organization is to accomplish objectives. A mission provides these objectives. Purpose is the understanding of an objective and the intent to realize it. An organization can be busy and

even over-worked, but unless all effort is purposeful the people are but a collection of wanderers. Objectives in organization must override efficiency, for efficiency is never an end in itself. Perhaps the principle is best illustrated by the tale of the organization-and-methods examiner who, after attending a symphony concert, reported that all 12 first violins were playing identical notes and the staff of this section could be drastically cut, that the oboe players had peaks of activity and their work should be spread more evenly, and that excessive refinement was observed in playing sixteenth and so-called "grace notes" which should be rounded off to the nearest eighth note. If, as opined by the conductor, there might be a falling off in attendance, it should be possible to close some sections of the auditorium entirely with consequent savings in janitor service and other overhead.

The principal objectives of the Bureau of Ships are to serve and support the line of the United States Navy, to act inside and outside of government as our Navy's technical agent for shipwork, and to manage creatively and economically all things that go into the life cycle of Naval ships from preliminary design to scrapping. Detailing and dissemination of objectives is a continuous process through daily meetings of the Chief with his top management, a weekly Chief's Planning Council, manuals and directives, and a monthly Message from the Chief in the *Bureau of Ships Journal*. Objectives determine the structure of an organization. The Bureau of Ordnance develops weapon systems and is organized around that objective. The Bureau of Ships develops the components, processes, and designs for ships and then builds and maintains them, and so must be organized for all these purposes.

Coordination is the combining of individual effort to achieve the organizational objectives. Work is divided because of time, space and specialization. Work division is the reason for organization, and the necessity for coordination. Coordination appreciates, for example, that research is best done by the users if early application of the results of specialization is sought. It recognizes that special technical skills and close coordination are not always compatible (4). Coordination seeks the reciprocal relating of all factors in a situation. It is a concert of endeavor (24).

In the Bureau of Ships, coordination is practiced by clear delegations of authority in the vertical direction, and by indoctrination in the common purposes in the horizontal direction. The Ship Type branches coordinate the contributing Technical codes, the Shipbuilding Assistant coordinates the new construction effort, the Comptroller coordinates the budget preparation, Design coordinates contract design specifications, Field Activities coordinates the shipyards, and Research coordinates the laboratories. This endless fabric of coordination is a prime consideration in evaluating the organizational pattern of the Bureau of Ships.

ORGANIZATIONAL PATTERNS

A functional division of work, which goes back to the classic studies of Frederick Winslow Taylor (32), has proved to be the best organization for technical progress. Able to comprehend less and less of the sum total of human knowledge, men have learned to specialize highly and to work together in specialized groups. Instead of giving his attention to all matters in one operating element, a man devotes his attention to one matter in all operating elements. Our laboratories underscore this organizational principle, and the continued preeminence of the Bureau of Ships in naval architecture, marine engineering and electronics depends upon it. Let us call this the horizontal directional of organization, Figure 7, as exemplified by the Technical branches, such as Boilers, Communications, Sonar, Radar, Gas Turbines, Damage Control, Electrical, and Minesweeping, under the Assistant Chief for Technical Logistics.

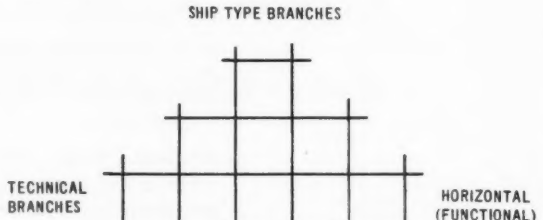

Figure 7. Horizontal pattern for functional expertness; Vertical pattern for administration and production.

But a non-functional or purpose organization, cutting orthogonally across the functional pattern, has proved superior for administration and production. Here we recognize Ship Type branches, such as Submarines, Destroyers, Auxiliaries, and Aircraft Carriers, running in the vertical direction across the Technical branches, which provide the expert functional support, Figure 7. Inevitably, the various scientific and engineering disciplines which go to produce a ship must be brought to focus on that particular ship to design, to build and to maintain it. In that sense, all of the functional organizational segments under the Assistant Chief for Technical Logistics support the purpose-oriented organizational segments under the Assistant Chiefs for Ship Design and Research, for Shipbuilding and Fleet Maintenance, and for Field Activities, Figure 8. Extending the pattern, such service segments as

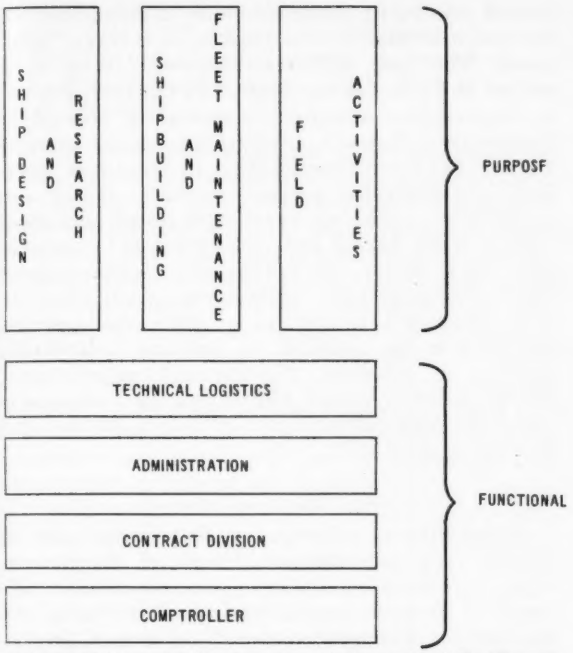

Figure 8. Organizational posture of elements of the Bureau of Ships.

the Comptroller, the Director of Contracts, and the Assistant Chief for Administration are also functional and have the same organizational posture as Technical Logistics.

The Bureau of Ships and most of its field activities therefore have a composite horizontal (functional) and vertical (purpose) corporate structure, and are generally classed as functional organizations. The pattern is essential for optimum specialization and technical progress but, at the same time, is difficult to coordinate. Most people are instinctively do-it-yourselfers and like to control directly all the elements in and supporting their job. It is also more difficult to keep a balanced workload in a functional organization because of the complex interrelationships. However, the functional-purpose pattern definitely appears best for accomplishing large volumes of detailed engineering and production work.

A line-staff organization retains the vital versatility and flexibility of a line organization, while benefitting from the specialized proficiency and expertness of a staff. A staff counsels; the line commands (29). A staff officer exerts his influence through the authority of his ideas and the acceptability of his work. A staff is something to lean on.

A functional-purpose organization like the Bureau of Ships actually has a format somewhat parallel to a line-staff organization like Cape Canaveral, Figure 9. The Chief of Bureau might be said to have his Special Staff in such assistants as:

Inspector General
Administrative Aide

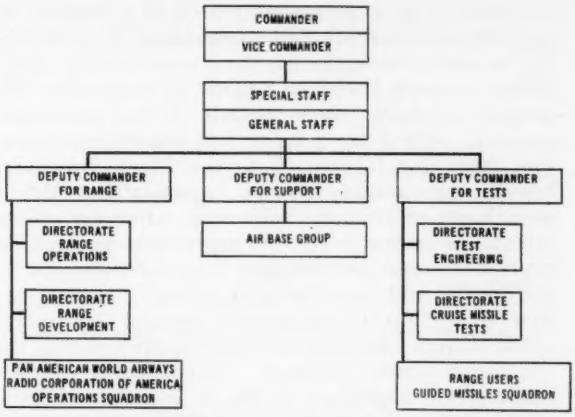

Figure 9. Line-staff organization at the Air Force Missile Test Center, Cape Canaveral.

Director of Planning
Coordinator of Undersea Warfare
Director of Value Engineering
Legislation and Liaison
Counsel
Patent Counsel

and his General Staff in the organizational segments:

Administration
Comptroller
Contracts
Technical Logistics

all supporting the Chief and the substantive operations of the Bureau:

Ship Research
Ship Design
Shipbuilding
Fleet Maintenance
Field Activities

Lower echelons of staff support in the Bureau, such as the Project Coordinators in Design and the Program Assistants in the Electronics division, are manifestations of the directorate staff concept as it appears in some line-staff organizations.

The bureaus and offices of the Navy Department generally constitute functional organizations, whereas the operating forces are line and staff organizations. However, the line and staff designations are sometimes applied to the bureaus of the Navy Department, meaning usually those headed by a line officer or a staff corps officer. The line bureaus are broadly those that produce the Navy's vehicles and their weapons, and the skilled personnel to man them. Categorizing another way, the Bureaus of Aeronautics, Yards and Docks, Ordnance and Ships are often referred to as the material or technical bureaus, while the Bureaus of Naval Personnel, Supplies and Accounts, and Medicine and Surgery are those that provide the general Navy-wide staff services.

Projectizing is a task force type of organization. It collects or coordinates the particular talents and

facilities to do a particular job. It is a vertical or purpose organization. The advantages of projectizing include getting the job done more quickly, providing a single leader and point of contact for the project, shedding non-essentials to the particular program, selling the project, and expediting transition from the laboratory to the Fleet. However, heavily projectized internal organizations would soon begin to compete with each other for scarce talents, to create their own supporting services, to overinflate programs because they enjoyed priority and funds, and eventually to swing from progressive leadership to stagnation because functional specialization had been sacrificed to projectizing. As the Congressional Advisory Committee on Undersea Warfare expressed it, "there is a distinct limit to the number of vertical organizations which can be operated at any one time, and the value of this system quickly declines once this number is exceeded" (33).

The success of projectizing is highly dependent on leadership, particularly when the technique is used as an overlay in a large functional organization like the Bureau of Ships. For a number of years Electronics was essentially projectized in the Bureau with its own organization and even many supporting services. Here was a young, fast-growing technology of extreme importance to the future of the Navy and the Nation, and projectizing was most appropriate to give it focus and momentum. Nuclear Power is another area of similar importance that has been projectized with similarly outstanding success. The most recent example is the *Polaris* program, in which the Director of Special Projects coordinates 125 government and private activities toward the job of producing a weapons system composed of nuclear-powered submarines to launch Fleet Ballistic Missiles. In addition to formal projectizing at the Assistant Chief level, there is a great deal of natural projectizing throughout the Bureau no matter how the organization chart is drawn.

Boards and committees are sometimes used loosely in an effort to shore up an organizational pattern. When someone mentioned to Karl Compton how remarkable it was that Lindberg could fly the Atlantic alone, he is reported to have replied, "It would have been still more remarkable if he had done it with a committee." According to Ralph Cordiner, "A committee moves at the speed of its least informed member and too often is used as a way of sharing responsibilitiy." Committees recommend and advise. "Individuals act while groups debate," writes Catheryn Sechler-Hudson (5). An exception to this pattern is a committee of line officials who have the power to commit and implement as, for example, the Joint Chiefs of Staff.

Evolution is a necessity in any human organization, and has been continuous in the history of the Bureau of Ships. Anticipation of organizational adjustments to developments in technology and to changing demands and objectives is properly an area of continuing concern to alert management. To express it as did the Commander of a Naval Shipyard, "The only difference between being in a groove and a rut is the depth." In fact, the Bureau of Engineering actually maintained a permanent Organization Board to study all proposed changes and to make recommendations to the Chief (34). Able and dedicated people can make almost any organization work, but these same people can often make a more logical and cleanly-tailored organization work better. Unfortunately, reorganizations have sometimes been undertaken in an effort to solve personnel, procedural or other management problems better handled by treating individuals, policies or practices. Change and improvement should not be confused. Most times the weakness is not in the organizational pattern but in the ties that bind the organization.

THE TIES THAT BIND

Organizational principles dealing with span of control, unity of command, delegation of responsibility and authority, synthesis of accountability, networks of communications, methods of working, and distribution of resources—these are some of the ties that knit organizations into whole and useful cloth.

Span of control is a function of organizational stability, diversification and distance. A supervisor is expected to give each subordinate direct attention as well as to coordinate their efforts with each other, thus augmenting his channels geometrically rather than arithmetically, Figure 10. On the other

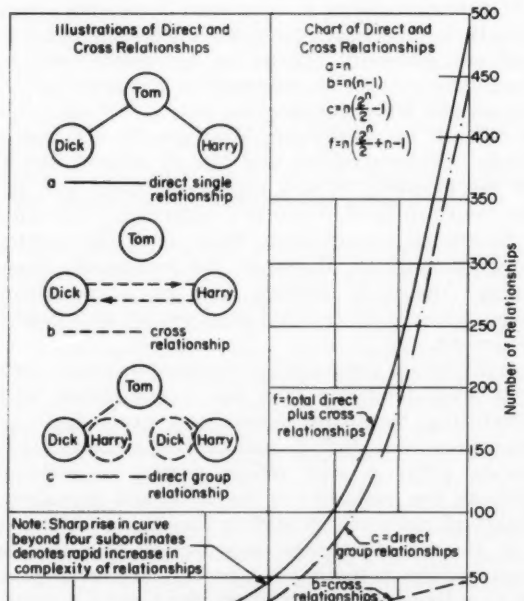

Figure 10. Graicunas Law of Span of Control.

hand, the best organizations often have the fewest possible stages of delegation. Clausewitz pointed out that "an order loses in rapidity, force and exactness if the graduation ladder down which it has to descend is long."

In the Bureau of Ships, the Chief has reporting to him 7 Assistant Chiefs of Bureau, 2 Independent Directors, and 8 Special Assistants yet the span of control is unbreached. This gross violation of the Graicunas Law (35) has proved practical in the Bureau of Ships because (a) the Chief is invariably a man who knows in detail every facet of the organization and operation in and outside of the Bureau, (b) he has a Deputy and Assistant Chief who is truly a management alter ego, and (c) his 6 other Assistant Chiefs have been delegated full authority to act for him in their respective areas. Below the Assistant Chief level, the span of control reverts more nearly to normal and extends usually from 2 to 8 subordinates, except where a directorate type of staff organization is used. To illustrate an extreme span, 19 technical desks, back in October 1942, all reported administratively to the Construction branch of the Shipbuilding division. The stages of delegation in the Bureau comprise the following principal operating levels:

Chief and Deputy	Policy
Assistant Chief	Review
Division	Coordination
Branch	Working

Unity of command contemplates that a man should be required to serve only one master, that humans need to hear a dominating voice in times of confusion, that groups are not adapted to effective supervision, and that no single position in an organization should be subject to definite orders from more than one source. In principle, no organization can long continue to operate effectively under a dual command system, any more than the Russian Army could wage a counter-offensive after Stalingrad with both officers and commissars in joint command. In fact, there are numerous acceptable exceptions, particularly in government. Many of these exceptions are due to the widespread functional environment in which naval engineers work and to the specialized services of the commands which they lead.

The Chief of the Bureau of Ships has in reality a constellation of bosses overhead. A bureau chief is in such demand outside that he may even find it relaxing when he has an opportunity to work with his own organization. His position is at the apex of two pyramids, one of them inverted. Multiplicity of overhead command occurs widely in the Bureau's field activities, and in specialized staff work, Figure 11. In most of World War II, Ship-Technical and Material reported to both the Director of Shipbuilding and the Director of Ship Maintenance. The Bureau of Ships internal organization is now largely purified of situations of this kind.

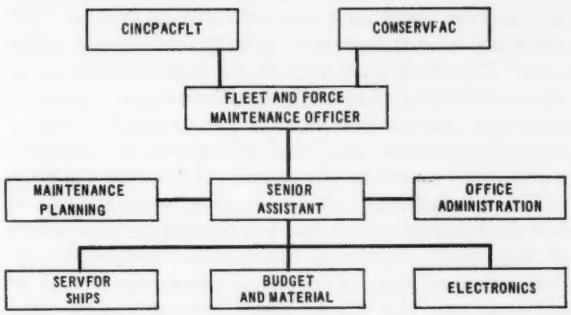

Figure 11. Dual Masters of the Pacific Fleet Maintenance Officer.

The fundamental responsibility of the head of an organization is to accomplish his mission. The purpose of delegation is to extend personal capacity, but delegation can never shift or avoid responsibility. Delegation means that the subordinate becomes responsible for doing the assigned job; his organizational senior still remains responsible for getting the job done. Generally, less important duties are delegated, while the more important duties are reserved. Clear understanding of responsibility for each area of work, both in scope and in nature, is essential to effective work subdivision. The acceptance of responsibilty carries with it the obligation for its performance. And so, through successive echelons, an organization develops the structure to accept and take responsibility. The equating of responsibility and the capacity of individuals is one of the neat tasks of leadership.

The Chief of the Bureau of Ships assumes complete responsibility for his organization. All official letters emanate from the Bureau under the title of the Chief. The privilege of signing letters and the individual responsibility are delegated but not the overall responsibility. On the other hand, some lack of clearcut inter-bureau responsibility has been a frequent criticism of the Navy's bureau system. Diffused fiscal responsibility and accountability led in 1951 to the performance budget.

Authority is permission to act. It is the partner of responsibility, and the teeth in coordination. A characteristic of excellent management in large private firms is an organization with an uncluttered flow of authority (36). Delegation of authority in the same ratio as responsibility is a principle to be sought, but in government rarely achieved. Our seniors often appear to have more authority than they actually do, and even the Chief of an organization is hemmed in by qualifications and exceptions to his authority. Authority like strategic weapons is often more effective if not used. "In pol-

icy making," said Secretary of Defense Wilson, "the arbitrary use of authority usually indicates a weak case." The best authority, in or out of uniform, is still proficiency. Authority legitimizes an organization, and makes decentralization possible. Organizational structure and the delegation of authority are at least as old as Moses who, following the sage advice of Jethro, ". . . chose able men out of all Israel, and made them heads over the people, rulers of thousands, rulers of hundreds, rulers of fifties, and rulers of tens. And they judged the people at all seasons: the hard cases they brought unto Moses, but every small matter they judged themselves," Exodus XVIII, 25 and 26.

The Bureau of Ships has made significant progress in equating authority and responsibility, as for example in the delegation to the field activities of most Civil Service classification actions, authority to approve technical conference travel, and greater fiscal authority. Dangling executives who have no authority have been avoided in the Bureau. Serious organizational inequities, as occurred when the Bureau of Construction and Repair accepted responsibility for ship stability but did not have commensurate authority to enforce it, are now largely extinct.

Accountability means answering for what has happened to the responsibility and authority that have been delegated. The accountability principle in organization has been growing in emphasis in the past decade, and standards and measures of physical and financial accountability have been markedly improved.

In government, Congress giveth and Congress can taketh away. Overall there is a strict accountability to Congress through the budget and report process, through audits by the General Accounting Office, and through the investigative prerogative. In and under the Bureau of Ships, management accounting is being developed to integrate planning, budgeting, programming, progressing and financial accounting in a system to reflect comprehensively for a commanding officer a continuing measure of how well he is combining the elements of personnel, material and objectives. Job standards are being developed, a leadingman is more strictly accountable for his men and their working time, and field activities are more susceptible to management analysis.

Internal communications convey management decisions and needs downwards and the action and advice of the organization upwards. This process produces the collective brain which symbolizes an organization with free-flowing channels of communication. In fact, a test of management is how readily it can obtain consultation from different parts of the organization. Among the factors influencing communications are physical proximity, work habits, extent of specialization, and managerial climate. Informal communication channels are equally as important in an organization as the formal lines following the chain of responsibility, authority and accountability. Daily business is done, regardless of the organization chart, according to a subtle and unrecorded communication network. These channels span horizontally every level of the organization, and knowing is the manager who can tune in on these loaded frequencies.

With its multitude of complex and interrelated operations, communication in the Bureau of Ships is a way of life. Shortening and assisting lines of communication was a major reason for the reorganization of 1958. Outgoing action zigzags upwards between the horizontal spread of specialties and vertical echelons of responsibilities. This movement has been typified by the communication network between Design and Ship-Technical, and took on added importance with the establishment of Technical Logistics. The Bureau of Ships route sheets and the comments thereon become prime internal communication documents of the organization.

The national economy has been likened to a pyramid with industry as the base, and the Navy at an important and rewarding level near the apex. Hence, the objectives of the Bureau of Ships can be achieved only by operating in a network of national industrial relationships. The pattern of multiple outside contacts interpenetrating the national economy is repeated in the 11 Naval Shipyards, 7 Naval Laboratories, 7 Supervisors of Shipbuilding, and 14 Industrial Managers under the management control of the Bureau of Ships. The extent becomes international in scope because the technologies contributing to ship design are world-wide. The foreign shipbuilding liaison organization of the Bureau of Ships comprised offices in Paris, the Hague, Rome, London, Lisbon, and Madrid responsible for a 248-ship program across 11 countries.

External communications of the Bureau of Ships therefore require a large number of contact points with the outside. These contact points should be easily recognizable, both from the outside looking in and from the inside looking out. For example, Design is the Bureau's front office to the Ship Characteristics Board, the Technical branches to the suppliers of components, Field Activities to organized personnel groups, and Shore Electronics to world-wide communication and navigation systems, Figure 12. Crossing the inter-face monthly between the Bureau of Ships and Outside are approximately 8,000 incoming and 3,000 outgoing naval messages, 130,000 incoming and 120,000 outgoing letters, and 1400 outgoing long distance calls. The Bureau of Ships is represented on over 200 outside committees and chairs 28 of them. Representatives of the Bureau travel some 9400 times yearly for first-hand personal contact with this far-flung shipbuilding and maintenance complex. The multifarious channels of outside communication provide also for feed-

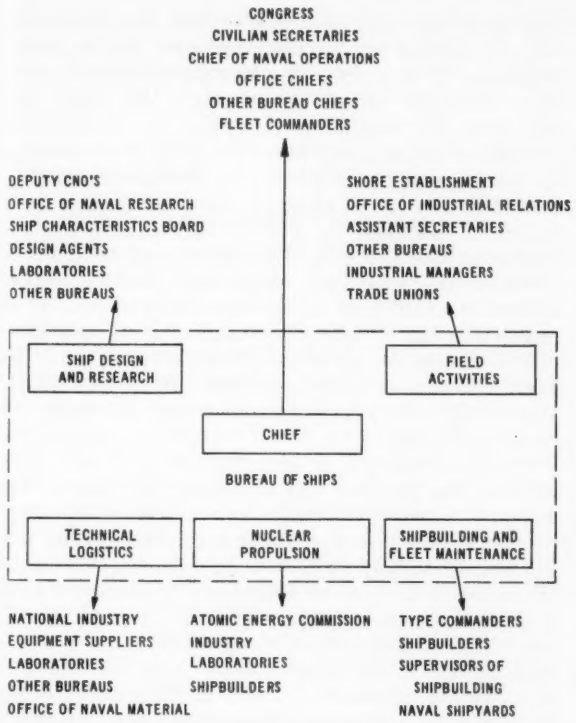

Figure 12. The Bureau of Ships is a highly extroverted organization with thousands of outside contacts hourly.

back, an essential need of a dynamic organization.

Method is a way of working. Broadly, the major methods and processes of organization and management are considered to be:

Policy Making
Planning
Structuring
Staffing
Directing
Coordinating
Controlling
Budgeting
Reporting

This is the gear train of organization in action. Methods condition the results obtained. The Bureau's budget is no better than the planning input. Research in the Bureau's laboratories can be no better than the quality and validity of the procedures. The ability to analyze an organizational objective and to determine the suitable method of attainment is a true test of a manager.

Resources for national defense are limited in war by physical availability, and in peace by the Nation's economy. The budget process is the effort to divide equably the available resources. Successive levels of review have increasingly broader view of relative need and are in a successively better position to evaluate and balance the division. At the apex, it is the function of the Congress to divide the expendable portion of the national income.

Men, money and materials are the fuel to make an organization run. The Bureau of Ships self-disciplines itself and its field activities in these resources through distribution of personnel ceiling, obligational authority, and facilities approval. Decisions in this area are truly decisive, and doubtless are the most critical tie that binds organization. Effective stewardship of these resources depends upon organization men.

ORGANIZATION MEN

Monsieur Pierre Danel, long-time director of the SOGREAH laboratories at Grenoble, once remarked to me: "One does not build a laboratory. He builds a staff." And so it is with any organization. In most organizations nowadays, primary concern of management is no longer with structure and procedures per se but with individuals, no longer with studying charts but with studying people. It is not the facilities nor procedures nor charts but the individuals who do the work. Many individual skills are vital in a large organization, but a highly technical organization like the Bureau of Ships depends basically upon its engineering specialties, officer and civilian, to accomplish its mission.

In principle, the entire Defense Organization, with the exception of the Secretary himself and the Under Secretary, is an organization of specialists. Serving our Nation, the Navy specializes in war at sea. Serving the Navy, the unrestricted line of the Navy specializes in command and operations at sea. Serving the Line, Engineering Duty officers specialize in naval engineering. Serving Naval Engineering are hundreds of specialized civilian areas each with its experts. Specialization is not a matter of fact but of degree. As General Nathan F. Twining remarked, "It is so difficult to get the whole truth these days because everybody is an expert."

Since Act of Congress in 1878 authorized the detail of naval officers to scientific schools or colleges for the purpose of promoting "a knowledge of steam engineering and iron shipbuilding," many naval engineers have been technical leaders not only of the Navy but widely in the nation. Officers receive post-graduate education and are designated for Engineering Duty in the Navy's technical organization:

a. To supervise research, development, design, construction, and maintenance of naval ships,
b. To permit the Navy to retain technical control of its own organization,
c. To form the bridge with national industry and with the scientific fields contributing to naval design and shipbuilding, and
d. To facilitate and insure full responsiveness of naval engineers in and out of government to the needs of the operating forces.

Vice Admiral E. L. Cochrane has said that learning ship design is a life-time job, and so are many areas of specializations. The Navy needs submarine, aviation, supply, medical, operations, and technical specialists, complementing each other towards the Navy's mission. To put the future of our Navy in any area of specialization in the hands of quasi-experts internally and private interests externally is simply not good enough. The challenge is to recognize that the Navy of the future is a Navy of pros covering different areas and degrees of specialization, and to lead this skilled team with understanding and inspiration. Admiral E. J. King was a strong proponent of this principle (37).

Naval Engineers are considered to include also specialized civilian areas as electronics engineers, marine engineers, physicists, mathematicians, electrical engineers, chemical engineers, metallurgists, ordnance engineers, nuclear engineers, structural engineers, aeronautical engineers, oceanographers, mechanical engineers, and naval architects whenever they are engaged in the art and science applied in the design, construction, maintenance and operation of naval ships and their installed equipment. The naval engineer has long ceased to work only with the tools of his particular scientific or engineering degree. In a mid-century of exploding technology, he has been thrust upstairs into organizational positions with broad responsibilities in controlling funds, facilities, and personnel. Most naval engineers now follow a personal career pattern in three broad and overlapping phases:

A. Formal study and basic experience.

B. Highly technical professional work.

C. Supervisory and management responsibilities.

Professor James N. Mosel of the George Washington University suggests that many naval engineers find it difficult to make the transition to the third phase because they continue to be scientifically-oriented rather than becoming more institutionally-oriented. A scientifically-oriented individual will expect a specific answer to a management case study when actually there may be no really good answer or dozens of them. The scientifically-oriented manager is like the "similarity" man in the classic satire of Joseph P. Maxfield who "tends to think first and act afterwards, if he ever makes up his mind to act at all"; whereas the institutionally-oriented supervisor resembles the "difference" man who "tends to act first and think afterwards, if he ever takes time to think at all." The key to any organizational effort is, of course, the coordinating ability of line supervisors. Mosel concludes that the abilities most essential to the coordination of technical organizations are (a) leadership, (b) administrative, and (c) technical, in that order of relative importance. These are optimized in the thoroughly able professional engineer or scientist who learns and practices overt orientation toward the organization of which he is a part.

People, like plants, grow or die. The health of any large technical organization can be maintained only by on the job, in the classroom, and at home education of its people. The Bureau of Ships is knee deep in talent, officer and civilian, but stays that way only by continuous attention to professional growth of the organization. New engineers coming to the Bureau of Ships receive formal training of from 6 to 17 months duration, depending upon their technical specialty, as well as time at appropriate field activities of the Bureau. Opportunities are provided all the way from cooperative undergraduate education, half time in the Bureau and half at a university, up to advanced management and doctorate studies. In the Naval Shipyards, as many as 4,000 different courses have been given in a single year, and total formal instruction averages 24 hours/man/year.

Frequent changes in duty station were once considered the peculiar and unenvied practice of the military services. Now the advantages of mobility in management and engineering development are recognized widely, and roughly three-quarters of all moves in the United States are accounted for by members of large organizations (3). Unquestionably one of the strongest features of the Bureau of Ships organization is the input arising from the mobility of its officer personnel. First, the ED officers bring wide technical experience and knowledge from duty at sea, at institutions of advanced education, in the shipyards, in laboratories, and in dozens of other field activities. Secondly, the officer system provides naval engineers having the closest possible ties with Navy Line operators, and prompt understanding, translation and feeding of their needs to the Navy's supporting technical organization. It is doubtful that any private technical organization has a better executive development procedure or a sales force any better fitted to recognize customer reactions and needs.

It has been said that bureau organizations are set up to do large amounts of routine work, and that there is little room or environment for creative work. The Bureau of Ships manages to do both. Surely few processes are more creative than starting with a blank sheet of drawing paper and developing ultimately the most comprehensive, shapely, complex and near-living engineering organism produced by man—a naval ship. Beyond a certain point, creative work cannot be expedited by increasing the manpower, any more than the time to boil a 2-minute egg can be halved by doubling the number of cooks. Original ideas are still more fragile than men—and fewer in number. Often in a technical organization only a very few do any real creating. Even in a laboratory the number may be as low as 1 man with 5 to 15 professionals around him. Yet in every technical organization, there are tremendous unknown and untapped reservoirs of undeveloped personal potential. The Bureau of Ships is so nurturing creativity that some of the shapes of ships to come look truly out of this

world (38). An Advance Studies section was established whose sole purpose is to look out into the blue. Every area of the Bureau is given tangible encouragement and assistance in new ideas. The laboratories are given the opportunity for self-generated projects. A creative climate is one criterion of organizational maturity.

CRITERIA OF ORGANIZATIONAL MATURITY

The mature organization, like the mature individual, can absorb impacts. It is a matter of organizational character, like dignity, loyalty, integrity, readiness to help and readiness to accept responsibility. Organizations which have the least outside assistance and the toughest problems often develop the strongest character. Resiliency in organization does not mean continuity and flywheel effect to the point of over-conservatism. It does mean the attitude and ability to drive immediately to the core of technical difficulty, to admit mistakes fully and sincerely, and then to bound back with vigor and insight that turns technical adversity into another long stride forward. The psychological studies of the Overstreets (39) have led them to place resiliency as the key criterion to maturity in an individual; perhaps it is also near the top of the list in evaluating an organization. The chief of an organization, unconsciously perhaps, diffuses such qualities by making his organization a part of himself and of his philosophy.

Mature organizations have developed adequate structural and procedural relationships for the prompt flow of information. Large organizations like the Department of Defense tend to acquire overlays and appendages which subsist on information provided by the productive components. Such productive components include the Bureau of Ships and its field activities, which are prime furnishers of information on technical subjects from billion dollar shipbuilding programs to millidollar cotter pins in any degree of detail desired. Responsiveness is engendered by Engineering Duty officers in the Bureau who comprehend quickly because of their background the information desired and who supplement specialized answers with balancing information from their own experience. The mechanics of information gathering is facilitated by the Bureau's functional-purpose organizational structure which designates specific individuals as know-it-alls in their area.

Like the checks and balances in the Legislative branch, the Executive branch, and the Judicial branch of our Nation's organization, so the existence of checks and balances in the Bureau of Ships is basically a process of producing organizational action and of reaching an organizational judgment. Checks and balances are actually the relating of all factors, considerations and expertness bearing on an organizational decision. The process of developing contract plans and specifications for new ship designs is a process of checks and balances. Though sometimes frustrating to officers with advance-base concepts of bureau management, such technical staffing is a mark of maturity and is usually essential to establish the soundness of decisions which have broad implications.

Centralization in organization is a relatively easy direction of movement. Decentralization requires maturity in knowledge of the operation, confidence in subordinates, and willingness to take calculated risks; the gains are great, and well-appreciated by private industry. The Bureau of Ships has been particularly successful in decentralizing widely both internally throughout a staff of 3700 military and civilian, and externally in more than 100 field activities with over 115,000 personnel. In the Bureau, each Assistant Chief and independent Division Director "is delegated great authority, leaving the Chief of the Bureau relatively free to handle matters transcending any one division" (4). In research and development, for example, responsibility for technical control of the Bureau's programs is decentralized internally to the users, such as the many specialist areas in Technical Logistics. Outside of the Bureau, the Chief decentralizes to the same extent to the commanding officers of the various field activities. Decentralization is facilitated by a rhythm of professional disciplines throughout the technical organization. For example, the reorganization of the Bureau in 1958 provided machinery, electronics, and hull professional backgrounds in each major ship subdivision. Such a pattern helps to strengthen the psychological relationships which most often form the main channels of communication.

At Emeloord on one of the amazing polders made from the Zuider Zee, there are the words: "A country to survive must build for the future." An organization builds for the future by planning. As in the Netherlands, planning is best done systematically with the total situation in mind. Decentralized planning, as in the Bureau of Ships, depends heavily upon each individual visualizing the position and importance of his task in the organization. It has been told that a civilian visitor aboard an aircraft carrier spoke to two firemen on watch in the boiler room. Upon inquiring what they were doing, from one he received the answer, "I'm tending the burners"; from the other the explanation, "I'm powering this ship." Planning in the Bureau of Ships follows the concept that the workers on day-by-day business have this larger understanding of the significance of their task and of the objectives of the organization, and so are in the best position to envision and plan the future in their areas of responsibility. Individuals are brought together psychologically when they are concerned not only in the work around them but also in the end use. Though decentralized in principle, planning in the Bureau of Ships has well-recognized coordinators. For example, a budget is simply the numerical expression of a plan in a common denominator, the only com-

mon denominator for personnel, material, and operations. The Comptroller is the Navigator for the Chief in the turbulent seas of budgeting and accounting, and thereby becomes one of the principal coordinators of planning in the Bureau of Ships.

Nowhere is it easier to waste money than in engineering. Small beginnings lead to large expenditures, and rarely is the cost of a new development underestimated. The mature organization visualizes ultimate possibilities and consequences, does not hesitate to give objective predictions and recommendations, and seeks always to reverse the trend formulated by Professor C. Northcote Parkinson (40).

A mature organization questions the premise that more employees means that more work will be done. Coordination is actually limited by such things as size, personalities, methods, knowledge, leadership, and wisdom, all of which point to better people with the appropriate qualities but fewer in number. Quantity is no substitute for quality. Number of employees and amount of work approach proportional increases only when emphasis is on production, not on organizational support. In the limit, there was the Pacific advanced base after Korea where the consuming function of most of the individuals thereon became to support each other.

On the other hand, contrary to the continued disrepute in which duplication is held, there are many cases in which it would be uneconomical to do otherwise. Such things as geography, the necessity for review, similar functions on different levels, and the plain necessity for division of labor are examples. The increase in number of shipyards doing submarine work has been healthy duplication. Naval engineers are subjected to strict duplication and are motivated by the fires of competition from the moment they enter the Naval Academy. Dispersal and duplication of small calculating machines adjacent to the engineers needing them is just as wise as centralization of large and expensive general purpose digital computers. Either way, duplication or pooling, must justify itself.

Clues as to an informal organization, which so often occurs regardless of the formal organization, can be found in the actions of its people. If possible, it is often wise to formalize an informal structure, rather than to force a change in behavior patterns. Packaged solutions should be examined with suspicion and not applied indiscriminately any more than should organizational principles; rarely does one find substitutes for dealing situationally with individual problems. When efforts are observed to keep an activity or organizational segment employed, here can be a clue to the need for surgery. One real self-audit can be worth a dozen outside experts who touch and go. Above all things, the mature organization recognizes that its most priceless assets are the integrity and judgment of its leadership.

LEADERSHIP IN ORGANIZATION

Leadership is the art of accomplishing missions through people. Hence organization is the instrument of leadership. Organizational objectives are accomplished by persuasive leadership, not by authoritative command. The most difficult and important task of a Bureau Chief or any Commanding Officer is not command, it is leadership. Writes Vice Admiral Smith, Chief of the Bureau of Naval Personnel, "Leadership principles are taught but a good leader acquires that quality only through practice. Such an individual by his personal example inspires others; by his actions instills a sense of responsibility in others; and by his administrative ability creates and maintains an efficient organization for the betterment of the Navy as a whole as well as for those individuals for whom he is responsible" (41). Barnard refers to leadership as "the quality of the behavior of individuals whereby they guide people or their activities in organized effort" (31).

In government organizations, especially in the technical areas, leadership requires a considerable amount of extroversion and facility in getting along with people to reach sound decisions. Essential is the ability to articulate in writing, in conversation, and in pressurized and unpredictable situations such as Congressional hearings. Decision-making by the leaders becomes a most intricate process involving simultaneously such considerations as budgets, legal limitations, military policies, civilian policies, pressure groups, the Congress, work balance, competence, and morale. The highest speed computer could never sweep through all the considerations the mind must cover in many day-to-day decisions required in the Bureau of Ships, such as in management of the Shipbuilding programs. Decisions in government are a multiple process requiring first the articulation of needs and objectives, and secondly the consideration of almost an infinite number of variables. "The fine art of decision" writes Chester Barnard, "consists in not deciding questions that are not now pertinent; in not deciding prematurely; in not making decisions that cannot be made effective; and in not making decisions others should make " (31).

"The larger the organization," said the President of General Motors, "the less top officials can concern themselves with details. If they were to follow such a course, operations would soon stop or decisions would be made in ignorance of the real problem." When the author visited a large commercial organization last year, he was introduced to the Vice President for Research out at the bench in the laboratory working on a new modular circuit. It was inspiring to see an executive with such obvious technical ability in his field, but meanwhile engineers were standing by neither working nor learning, and the executive himself was surely not working at being a vice president. Perhaps only a coin-

cidence but it is understood that the company was reorganized 5 months later. A luxury of government is cheap management. Paul Appleby has said, "There is more inadequacy in government because of the inability of officials to operate on their proper levels than from any other single cause" (42).

No executive ever has all the management answers. He who does has ceased to grow and progress. To continue to think about these things is the important aspect. There is no substitute for the critical evaluation of one's own personal experience. Leaders in a technical organization learn to have a nice balance between belief and skepticism, to visualize organizational objectives and the steps necessary to their attainment, to anticipate and comprehend the moment of passing from one phase of a situation to another, to suspend decisions until actually required, to develop the ability to say "no" and the readiness to say "yes," and to bring to bear the collective judgment and experience of the organization. True leadership does not hesitate to question and reconsider those things which have been accepted and taken for granted. Leadership is work, and plenty of it.

There is just no substitute for good leadership, and large organizations are more sensitive thereto than is generally realized. Sometimes a single new personality in the management of a large organization can have a reverberating influence on the work of the group. The informal organization is particularly quick to sense any insincerity or lack of conviction on the part of their leaders, and morale and teamwork invariably suffer. Only forthright, impartial, knowing, and dedicated leadership can hope to develop whole-hearted fellowship in an organization.

The most powerful anti-Parkinson bulwark is the willingness of the chief of an organization to take calculated risks. To cut back personnel in the face of predictions of catastrophe unless increases are granted, to eliminate ruthlessly garnished feedings at the public trough, and to respect individual integrity and responsibility more than control and policing procedures, these things may bring no kudos but they will bring respect, loyalty and savings. The new General Manager who said to his Foremen, "Now just tell me what you need, fellows, and I'll tell you why you don't," may have latched onto this principle a bit tightly but he had the idea. The governing criterion is whether the objective can still be accomplished and the organization live. Even a rationale suggests Kettering's observation that "logic is a means of going wrong with confidence." Like a ship design, a need must look right to be right. If it doesn't, the calculated risk principle may be the tough but effective answer. Such tactics require leadership of the highest order.

The organization of the Bureau of Ships reflects the philosophy of its present and previous Chiefs, and is really a composite of their thinking. The acme of personal leadership is to shape the development of an organization not only for the duties of the moment, but idealistically toward goals of the future which can be realized only by one's successor, a principle so necessary in the military rotation pattern. The acme of organizational leadership is organization by objectives.

ORGANIZATION BY OBJECTIVES

The idea of organization by aims or objectives is the most inspiring, and can be the most effective, of all organizational principles. It is derived from the concept that human beings, knowing and understanding the objectives of the group, will cooperate and coordinate freely and wholly toward achievement of these objectives with a minimum of formalized structure, methods and controls. The principle emphasizes dominating goals versus dominating directives, unity of purpose versus unity of command.

The principle is not inconsistent with the conventional evaluation of management as being hard or soft. It concerns simply a higher level of human behavior. Management by objectives is fully compatible with an organization recognized as a taut ship. In fact, the very essence of the taut ship is for each man to have and know his responsibilities and to live up to them as an individual.

The principle is the antithesis of the psychological work contract (43) in which an employee de-emphasizes human values in favor of material values, in which he sacrifices growth toward his true potentialities for wages and security, and in which he gives up creativeness for passiveness and noninvolvement. On the contrary, the principle places the emphasis, not on money, but on the human striving to grow, to develop, and to feel an inner sense of worth.

Functional systems of organizations, like mass production, tend to limit the scope of individuals. However, a central purpose and common understanding and attention can give meaning to specialized contributions of specialized individuals. It warms our human interests and satisfies our higher wants. Naval engineers, like doctors, dedicate their lives and time to a cause, and respond to such high motivation.

Dr. D. M. McGregor of the M.I.T. School of Industrial Management explains the principle by reminding us that a satisfied need is not a motivator of human behavior. "Man is a wanting animal. As soon as one of his needs is satisfied, another appears in its place. This process is unending" (44). He has arranged man's needs in order of importance:

Self-Fulfillment	(Creative, realize potentialities)
Ego	(Self-esteem, reputation)
Social	(Belonging, acceptance, friendship)
Safety	(Protection, uncertainty, unpredictableness)

Physiological (Hunger, rest, exercise, shelter)

The fact that modern organizations largely satisfy physiological and safety needs has shifted emphasis to social and perhaps to egoistic needs, for the baser wants are no longer adequate motivators of behavior. In mature and creative organizations like the Bureau of Ships and its field activities, emphasis has further extended to self-fulfillment, the highest form of human motivation. The organization cannot satisfy the needs of a man for self-fulfillment, but it can provide conditions such that he is encouraged and assisted in seeking such satisfactions for himself. "Only the management that has confidence in human capacities and is itself directed toward organizational objectives rather than the preservation of personal power can grasp the implications of this emerging theory" (44).

Organization by objectives and by personal attitude is believed to be the key to creative progress. It has been a cardinal principle applied throughout the last 4 years by Rear Admiral A. G. Mumma, Chief of the Bureau of Ships: "I believe that the key to good management is to instill in all hands a basic progressive attitude" (4). Again at a laboratory dedication he said, "It is an important aspect of our visionary future that each of us be challenged to look ahead and see how far we can possibly see." Such is the stuff of which our Navy of the future is made.

Naval engineers have a profession as exciting, essential and challenging as any in the world today. If we continue to organize and function with our eyes in the sky and our feet on the ground, nothing can surpass us on the sea.

CONCLUSIONS

This study of organizational history and principles has led to the conclusion that certain characteristics are essential in any parent organization for Naval Engineering, either now in the Bureau of Ships or in the future. They are:

1. A unified organization responsible for all technical and administrative matters concerned in the design, building, and maintenance of naval ships, together with the applied research and development in direct support thereof.
2. A chief who has constant access to the military side of the Navy for matters that concern fleet operations and logistics and, at the same time, access to the civilian side of the Navy for matters of policy and business administration.
3. Naval officers who have an unsurpassed technical competence, and a completely responsive understanding of fleet needs deeply rooted in personal naval background and experience.
4. A civilian staff of minimum size and highest quality of true professionals in every scientific and engineering discipline embraced by naval engineering, and in those vital supporting categories such as law, accounting, and personnel.
5. Management control and command of naval shipyards, naval laboratories, and other field activities required for the direct support of naval engineering afloat and ashore.
6. A vast but person-to-person network of technical understanding and collaboration with American industry, which is the base of ship component development and naval shipbuilding.
7. A functional-purpose pattern of organization which provides the framework for maximum technical specialization and progress and, at the same time, brings such specialization to sharp focus on all the ship production and maintenance needs of the Navy.
8. An organization which has responsibility and authority cleanly delegated in reasonably the same proportion and the trustees held fully accountable for performance, which has resiliency, mobility, information, checks and balances, unity, foresight, and all the characteristics of maturity, and which is stripped down to only those essentials unquestionably indispensable to the accomplishment of its mission.
9. A progressive attitude pervading the organization, conditions of operation so that the people can achieve their own personal goals best by directing their own efforts toward the organizational objectives, and leadership which can inspire these things by being highly versed and experienced in every technical and management aspect of the organization and particularly in human relations.
10. Above all, an organization wholly dedicated in every fiber to serving the men, the fleets, and the shore establishment of the Navy.

The titles and terms will change with time, but these fundamental needs will live as long as the United States Navy.

ACKNOWLEDGMENTS

Supplementing the acknowledgments appended to Part I of the paper, appreciation is expressed to Captains G. W. Bailey, H. P. Rumble, E. J. Fahy and R. R. Bradley, and to Mr. C. E. Wilkenloh for their ideas and collaboration on a Board To Analyse Electronics Areas Of The Bureau Of Ships, December 1958.

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In the erection of structures in Arctic bases, it is necessary that the foundations rest firmly on the perma-frost. Each summer the top four to six foot layer of ground thaws. Excavation to the perma-frost level, necessary for a solid footing, is accompanied by the hazard that the layer becomes unstable as the top layers of ground are removed. To avoid this hazard, it is the usual practice to refill excavations with a foundation pad of non-frost-susceptible material such as gravel. It is also necessary to provide for ventilation of the pad and insulation of the floors of structures which rest on it to prevent the building from melting its own foundations by heat conduction into the permafrost. In recent construction of NIKE sites, the necessity arose to place a portion of the structure below ground surface for ordnance safety reasons. This increased the depth of difficult and expensive excavation necessary to provide an adequate insulating pad. In order to avoid this, a scheme was adopted of using a refrigerated foundation system. A fully automatic refrigerating plant supplies coolant coils built into the foundations and provides the necessary barrier to transfer of heat from the structure into its frozen foundation.

—from THE MILITARY ENGINEER, March-April, 1959

Official U. S. Navy Photograph

Two of the Navy's newest guided missile ships are shown in this bow-on view. Built simultaneously in a graving dock at Puget Sound Naval Shipyard, the USS *Coontz* (DLG 9) and USS *King* (DLG 10) are shown just prior to launching.

Two of the Navy's newest guided missile ships are shown in this bow-on view. Built simultaneously in a graving dock at Puget Sound Naval Shipyard, the USS *Coontz* (DLG 9) and USS *King* (DLG 10) are shown just prior to launching.

"WELDING JOURNAL"

PRESSURE EQUIPMENT FOR LOW-TEMPERATURE SERVICE

ACKNOWLEDGEMENT

C. R. Soderberg, Jr., is the author of this article. It was published in the January 1959 Welding Research Supplement of the "Welding Journal."

INTRODUCTION

RECENT years have shown a marked increase in use of low-temperature equipment. In part this has been prompted by new chemical processes, such as ethylene plants which require boiling point separations involving temperatures of −150°F. Oxygen and liquid-nitrogen plants depend upon the same type of low-temperature separation and their use has increased. Another factor has been the decentralization of industries, with new and large plants being installed in northern Canada and other frigid areas.

Although code rules for services below −20°F have existed for some thirty years, it is only in the last fifteen years that there has been any general appreciation that ferritic steels are subject to low-energy or "brittle" failure at ambient temperature. The many ship failures of World War II, which led to the organization of the Committee on Ship Structure Design, initiated organized large-scale efforts on low-energy fracture. Similar experiences with storage tanks, pressure vessels and structures have also promoted investigations of this phenomenon. While the manifestations and circumstances leading to brittle fracture are reasonably well established, a serious lack of fundamental scientific knowledge still exists. For example, the modified Griffith theory of crack propagation, which shows that stress and crack length are dependent, cannot be made to check with tests on a plate under tension. A pattern does seem to be emerging which permits the definition of a transition temperature. Well below this temperature, brittle cracks can propagate easily at stress levels far below the yield point. Around the transition temperature, stress levels approximating the yield strength are required to propagate a crack, whereas at temperatures well above the transition temperature, brittle cracks cannot propagate.

Statements have been made at times that brittle fracture of steel is a new phenomenon. Shank [1] surveyed past failures and found instances, at least as far back as 1879, which, with reasonable certainty, were brittle failures. These failures seemed to start shortly after Bessemer steel replaced the existing wrought-iron construction. This transition to Bessemer steel and open-hearth steel took place during the period from 1860 to 1890. All of the brittle fractures studied seem to be characterized by low temperatures, low ductility and rapid cracking. They seem to occur during an application or change in load, or with a drop in temperature. Welded steel structures appear to be more prone to failure. Almost none of these failures involved structures which were stress relieved.

TESTING

A voluminous amount of literature exists today on the subject of brittle fracture and this field has been ably summarized by Parker [2] and Shank.[3]

There is considerable detailed knowledge about specific materials but, seemingly, there is no consistent test which has been used to evaluate all materials. Present-day tests fall into the following four categories:

Stress Level in Path of Advancing Crack

This type of testing was first started by Robertson in England [4] but was modified and continued by Feely and his associates [5] at the Standard Oil Development Company (SOD). The first step in their test procedure is to initiate a fine crack in the test specimen. In some instances the crack was a jeweler's saw cut and in others a brittle crack started by impact in a portion of the test specimens cooled with liquid nitrogen. This crack is stopped by a high temperature at another spot in the test specimen. The test plate is then loaded in a tensile test machine where the plate temperature is controlled and a tensile load applied. A projectile is then fired at a wedge previously inserted in the original crack, and the minimum tensile stress for crack propagation determined. Their results are essentially as shown by Figure 1 where a stress curve they obtained for an ASTM A-285 steel is reproduced. One of the interesting features of this test is that it appears to define a stress level above which a brittle crack can run, and below which a crack cannot run. This result implies that it might be possible to design below the transition temperature without risk of brittle fracture if the stress level is kept low. So far it seems impossible to correlate Feely's work with Griffith's crack *length* theory.

Orowan [6] has pointed out that it is probably not correct to consider the plateau stress found in Feely's test as a material property, and that the plateau stress should be a function of crack length. Orowan, who has made some of the best fundamental contributions in this field, extended the original Griffith-Irwin theory [7] and arrives at the following expression for a brittle propagation stress:

$$\sigma = \sqrt{EP/C}$$

where:

σ = stress.
E = modulus of elasticity.
C = crack length.

P corresponds to a surface tension but applicable to ductile materials and is approximately equal to 10 lb/in. for carbon steel at atmospheric conditions.

If this expression is applied to Feely's test with a stress of 10,000 p.s.i., a crack length of some 3 in. is obtained. Feely varied the crack length in his tests from ¾ to 2 in., which should result in a change in stress of 60 per cent, but found *no* effect of crack length on his propagation stress. There may be some effect from the projectile load in the test plate but it is difficult to believe that this resolves the enigma. This discrepancy between Orowan's crack theory and Feely's tests represents one of the most basic problems which must be resolved before a clear understanding of brittle failure is obtained.

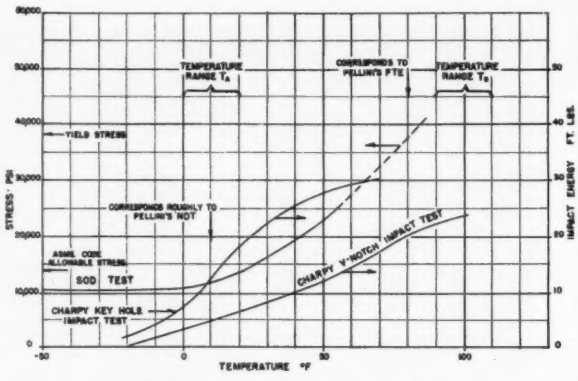

Figure 1. Charpy impact and SOD brittle tests on ASTM A-285 grade C steel.

The curve reproduced in Figure 1 is quite alarming because it indicates that a brittle crack can run in ASTM A-285 grade C steel at the ASME code allowable stress of 13,650 p.s.i. at about 20°F on the heats used in Feely's tests. The one reassuring fact is that Feely used a projectile with 2500 ft.-lb. of energy to start the existing crack running. If he reduced the energy of this projectile below 1500 ft.-lb., the required stress increased until at zero projectile energy the stress level had to be slightly above the yield stress. This indicates that the steel had a fair resistance to initiation of a running crack apart from its low resistance to the propagation of such a crack once started. Feely had a relatively large specimen—some 6-ft. long and 6 to 16 in. wide.

Fracture Appearance.

This group of tests defines a transition temperature by the *type* of failure which occurs. Usually 10, 20 or 50 per cent shear surface in the fracture is taken as the transition between ductile and brittle failure. In many respects this is probably the most desirable kind of test because it is simple and reproducible. Impact tests, notched-tensile tests and slow notch bend tests can be interpreted this way. However, the choice of the amount of shear to be used can and is subject to dispute.

Ductility.

A third group of tests is based upon the ability of steel to flow plastically before extensive cracking occurs. One example of this type of testing was evolved by Pellini and his associates at the Naval Research Laboratory. [8, 9] A weld overlay with low impact properties is deposited on his test specimen. He then uses either a drop-weight test or an explosion crack-starter test. The drop-weight test involves dropping a weight on a small specimen which is constrained so that it can only deflect 0.3 in. which is, however, sufficient to initiate cracking in the brittle weld deposit. The explosion crack-starter test consists of supporting a plate on a circular die and deforming this plate by an explosion. Pellini has

defined various types of transition temperatures which can be stated as:

Nil Ductility Transition (NDT)—Essentially no deformation is required for a fracture.

Fracture Transition for Elastic Loading (FTE)—Below this temperature fractures will propagate through elastically loaded sections, while above this temperature they will only propagate through plastically loaded parts.

Fracture Transition for Plastic Loading (FTP)—Brittle fractures are not able to propagate through the material even though the material is severely plastically deformed.

Another variation on this basic type of test is the Charpy specimen where the percentage of lateral contraction, rather than impact energy, is plotted versus the temperature.

Impact Energy.

This type of test probably represents the largest group of available tests. The results are reported as impact energy to cause failure of notched specimens versus temperature. The disturbing feature about this test is that small changes in geometry give an appreciable change in results. Two common types are the Charpy keyhole and the Charpy V-notch tests. The temperature difference between these two tests for a 15 ft.-lb. impact on mild carbon steels is commonly 30°F but may be as much as 50°F.

One item which is not really clear from any of these tests is how to separate the amount of energy required to initiate a crack from that required to make a crack run. Some tests are clearly propagation tests, such as Feely's, and in the author's opinion, Pellini's, but most of the others are a combination of crack starting and propagation.

One attempt to distinguish between crack starting and propagation has been made by Hartbower.[10] He used a two-blow varition of a Charpy test. His results indicate that, in the range of 15 ft.-lb. and lower, it takes considerably less energy to restart a crack and propagate it than it does to form initially the crack without propagating it. His transition temperature seem to be about the same for the first and second blows even though the energy levels are considerably different.

MATERIALS AND FABRICATION

One of the puzzling aspects of brittle fracture is the fact that austenitic steels exhibit little or no tendency toward brittle failure irrespective of the temperature, while all ferritic steels have a transition temperature below which the brittle strength is lower than the ultimate ductile strength. At present, it seems that only empirical knowledge exists as to the cause of brittleness, but some generalities can be stated.

A factor which has a powerful influence on transition temperatures is the heat treatment to which the material is subjected. Pellini quotes an instance[9] where a Ni-Mn steel was normalized and tempered with a resultant NDT of +60°F. A plate of the same size from the same heat was water quenched and tempered and the NDT dropped to −70°F.

In some instances it has been possible to relate this change in brittleness to ferrite grain size. Owen and his associates[11] described tests on three ship steels. By varying the heat treatment, they were able to change the number of ferrite grains per cubic millimeter from approximately three to six. In all three of the steels they tested this meant a drop in the Charpy 10 ft.-lb. transition temperature of 50 to 60°F. Another example of this phenomenon is described by Hartbower.[10] He tested an AISI 4340 steel which contained 1¾ per cent of nickel. With three types of heat treatment he was able to produce specimens with Rockwell C hardnesses varying from 27 to 45. The sample with a hardness of 27 had the smallest grain size and also the best transition temperature of −200°F (−130°C). The sample with a hardness of 45 showed a transition temperature of 77°F (25°C). These examples illustrated that heat treatment and its resultant grain size have a powerful influence on transition temperatures. In general, the finer the grain size the lower the transition temperature, although other microstructural changes may affect this. This is probably the explanation for the fact that production steels, which are supposedly made by similar procedures but vary somewhat in deoxidation and finishing temperatures, can show a wide variation in their NDT. Ship structure steels tested by Pellini showed a spread from 0 to 60°F.

Composition changes also show significant effects on brittle properties. Carbon content should be low and the manganese high for improved brittle properties at a given tensile strength. Boron, nitrogen and phosphorus seem to have harmful effects, whereas silicon, aluminum and nickel appear to have beneficial effects. Here again, alloying elements which are beneficial also decrease the grain size.

Since heat treatment has such a marked effect on brittle properties, the question concerning welds made in steels whose beneficial properties are induced by heat treatment is naturally raised. Pellini[8] has made a study of notch brittleness in welds. They behave roughly the same as a casting of the same composition. Any rules of chemical composition and heat treatment seem to apply as they do in the base material. Welds, however, because of minor flaws, do have a tendency to introduce cracks and, in the as-welded condition, high residual stresses add to the ease of initiating a crack. A weld in any sizable structure, however, is rapidly quenched compared to as-rolled plate with resultant grain refinement and reasonable brittle properties. Consequently, if normal care is exercised in choosing weld materials,

the welds and the heat-affected zone should not change any of the premises made for the base material as far as crack propagation is concerned. The influence of welding on the heat-affected zone of quenched-and-tempered plate is another matter and requires careful investigation.

One fabrication procedure which has caused several failures in the past is the initial hydrostatic test. For vessels which are to be used at elevated temperature, the transition temperature usually is not considered. However, during the hydrostatic test these vessels must withstand overloads under ambient conditions. To avoid risking failures, the vessel must be heated for the test or the material must be changed to resist brittle failure at ambient conditions.

Generally, fabrication involves considerable amounts of cold work and strain aging. This cold work can have a distressingly harmful effect on a steel's brittle properties, particularly its crack-initiation resistance. Gensamer and co-workers [2] cite examples where 5 per cent prestrain of ship steels raised the 10 ft.-lb. Charpy-V transition temperature about 40 to 60°F. It is not clear what the solution to this problem should be but, certainly, stress relieving is beneficial. The presence of residual stresses near welds, biaxial and even severely triaxial in nature, has been demonstrated to add materially to the ease of crack initiation when an external load is applied at or below T_A of Figure 1. At T_B of Figure 1 residual stresses can yield out by local plastic flow without adverse effect.

INADEQUACIES OF PRESENT KNOWLEDGE AND CORRELATION WITH SERVICE

In the previous section, the present state of knowledge was reviewed. In this section, the conclusions or questions the present information on brittle fracture implies will be considered. Figure 1 shows characteristic test results for a common steel. Two temperature ranges are marked on this curve. Temperature range T_B is at a point where Feely's brittle failure stress exceeds the material's yield point. At or above this temperature, which is 80°F for 1-in. thick plate, it is reasonably certain that A-285 will fail in a ductile fashion. This temperature is also on the upper plateau of both the Charpy keyhole and V-notch tests which are also plotted on this curve. At the other end of the curve, a temperature range T_A is shown which is at the start of the low-temperature plateau for Feely's tests. This temperature is roughly at the 15 ft.-lb. Charpy-keyhole value and about where Pellini's Nil Ductility Temperature occurs. Below temperature range T_A, the material has a high probability of failing in a brittle rather than a ductile fashion. The stress from Feely's test is 10,000 p.s.i. at the point which is uncomfortably close to the code-permissible stress of 13,750 p.s.i. for this material. The dilemma which prevails is that these two temperature ranges bracket the atmospheric conditions, and a sizable amount of construction exists in this range without a significant number of failures. A few brittle failures have, however, occurred just about at the midpoint of this range. The Fawley tank failures occurred at about 40°F with steel which had similar characteristics to that shown in Figure 1. The question which exists is the relative hazards of the two extremes of this transition range. At temperature T_B, it is reasonably certain that brittle failure *will not* occur, but one cannot be certain brittle failure *will* occur until reaching below temperature T_A.

Another enigma is the successful experiences with vessels using materials based on the Charpy keyhole 15 ft.-lb. acceptance test. No failures have been recorded where a material was impact tested at the mill and the steel was manufactured in accordance with a low-temperature specification. In light of present knowledge, it is extremely surprising that the record is as good as this when one considers that these tests are made only at specific locations in plates before fabrication and on arbitrary bars in forgings and castings. Present-day vessel construction, moreover, permits nozzles and other details with large stress concentrations and deplorably poor weld details. It is difficult to understand this experience record but it must be considered in any fundamental design decision.

In the study of ship failures, no plates in which fractures originated had Charpy V-notch impact value over 11.4 ft.-lb. and the average was 7 ft.-lb. at the failure temperature. Since these impact values correspond to more than 15 ft.-lb. in the keyhole tests, the better service record of vessels may reflect a lower order of restraint and, possibly, plastic flow at design or fabrication stress raisers. Vessels do not ordinarily contain stress raisers as severe as those found at sharp hatch covers.

There appears to be a higher incidence of failures on nonpressure storage tanks and on pressure vessels under test than on other structures. This seems to indicate that the average stress level and, equally or more important, plastic adjustment play a large role in determining the onset of brittle failure. The role of stress level is reasonably well demonstrated by Feely's tests but initial plastic-flow adjustment is still an unknown. These are probably part of the larger question of crack starting versus crack propagation. Choosing materials and operating conditions to limit crack *propagation* is certainly the safer alternative, but this implies T_B or above and rules out too much present-day satisfactory construction. Selecting material with reasonable resistance to crack *starting* implies T_A or above but puts a premium on adequately controlling stress concentrations, initial defects, residual stresses and fabrication tolerances. The relative hazards of points T_A and T_B are difficult to assess.

LIMITATIONS AND ECONOMICS OF PRESENTLY AVAILABLE FERROUS MATERIALS AND VESSEL FABRICATION

Before an attempt is made to formulate a design

basis, it might be well to examine present-day availability and suitability of various materials. Presumably the eventual solution to the brittle-fracture problem will be in the development of a test which more adequately determines the brittle tendencies of materials and specifications which economically control the manufacture of these materials. Though present-day tests and specifications exist, they are not entirely suitable or economical. It may, however, still be instructive to examine steel availability with these present-day tests and specifications.

In the United States, the carbon steels without any special specifications or other controls for brittle fracture are obviously the most available. However, they are unsatisfactory for the prevention of brittle fractures. Some carbon steels, with special specifications and controls which improve their transition temperature, are available, if the order is large enough to interest a mill. These impact-resistant steels involve an appreciable cost premium and presently cannot be bought on a heat basis. With a more widespread use, it should ultimately be possible to buy them in small quantities. Nickel steels and certain other alloys are also available with impact tests on mill orders but virtually impossible to get for any smaller quantities. Austenitic materials are available, although relatively expensive.

Europe, on the other hand, has shown a considerably more progressive attitude in solving this problem. Real effort has been shown by both steel companies and users in trying to make a carbon steel with improved brittle-fracture resistance. Included in their specification requirements are means of assessing the steel, usually by use of Charpy V-notch tests.

No real effort has been made in this country, except in industries associated with atomic energy, to fabricate structures so that welds and basic materials can be properly assessed. Blind-root partial-penetration welds are still permitted in any vessel nozzle operating in or below the transition range. This seems to be an open invitation to start a brittle crack. The economics of premium construction are still not clear. In general, building a vessel with an improved carbon steel and conventional fabrication details would seem to be less expensive than premium fabrication details and poorer materials. The former should also offer greater assurance against brittle fracture. Currently we have little choice, since the improved carbon steels are not commonly available. The first step toward improved steels would seem to be an open recognition of expected transition temperature ranges of present steels including the effect of chemistry and heat-treatment controls. New steel specifications could then be established in a similar fashion to the present practice on ship steels.

SUGGESTED APPROACH FOR AVOIDANCE OF BRITTLE FAILURE

This section will examine in more detail the limitations which must be placed on design, materials and fabrication to assure an economic structure reasonably safe from brittle failure. Three factors are necessary in order to produce a brittle failure in any steel structure: (1) an initial flaw, (2) a critical level of average stress high enough to produce plastic flow at the flaw location and (3) temperatures around or below the transition temperature. If any of these three conditions are not present, brittle failure cannot occur, which is part of the reason why such a small fraction of all structures built have failed in a brittle fashion. In order to formulate a criterion to minimize or eliminate brittle failures, each of these conditions will now be examined to see if it can be properly controlled.

With regard to elimination of initial flaws or incipient cracks, it is extremely difficult to assure the almost perfect construction and design which this would require. Every vessel which is built has some attachments such as nozzles or clips. Almost universally these are attached by welding and the present permisssible code details, which include blind-root and partial-penetration welds, make it almost impossible not to have some small flaws. Undercutting at the toe of welds is practically unavoidable especially at fillet welds. Material sensitivity to fabrication cold work and strain aging can affect the material at any of these discontinuities. With residual thermal stress, any of these discontinuities becomes a potential flaw. In consideration of all these factors, it must be concluded that the present code details and welding are not adequate to prevent the start of a brittle crack even though the critical amount of plastic flow required to start a crack is still unknown. This means that better methods of assessing welds must be used and present nozzle details refined to prevent the stress concentration presently permitted.

The next item which could be controlled would be the average stress level. Presumably from the SOD tests, if the stress level were kept substantially below 10,000 p.s.i. for A-285, a brittle crack could not run. One difficulty with this concept is that there is a definite uncertainty about the levelness of this low-temperature plateau. Local stresses, which can persist around the entire circumference of a vessel, can be considerably higher than the general stress level in present code allowable designs and, therefore, must be adequately considered.

Knuckles in toriconical heads are a good example, since in this case no accurate solution is presently available for the stresses. Moreover, it is probably more expensive to limit this kind of local stress to extremely low values than it would be to find a material with satisfactory impact properties. Local stresses must, therefore, be carefully controlled to eliminate regions favorable to brittle-crack initiation. This leaves the control of a material's transition temperature as the most important variable in eliminating or minimizing this type of failure.

In order to control the transition temperature it

must be defined more precisely. On the one hand is a mass of laboratory tests which indicates that, for maximum assurance, one should not design below T_B (Figure 1) of a steel. On the other hand, an equally large amount of service experience indicates that adequate service can be expected statistically by operating above T_A for any given material. There seems to be almost complete agreement that no structure should operate below T_A unless the general stress level is severely limited to the order of 3000 to 4000 p.s.i. There are inherent dangers in operating between T_A and T_B because a structure's safety depends upon a brittle crack not starting rather than the fact that a crack once started will not run as is the case above T_B. At the present time, it seems unwise to advocate so radical a step as to require temperature range T_B, since this cannot be supported by present experience. The lower transition temperature (T_A) must continue to be used as an interim step until a better understanding of the basic failure mechanism is found or until an inexpensive available carbon steel is found where temperature range T_B is well below 0°F.

With this background, the requirements can now be presented for an acceptable interim design basis for the use of ferritic steels at low temperature. First, a consistent, simple method of measuring the temperature T_A for a given material must be found. Present ASME code practice is to use the Charpy keyhole test and say that the transition temperature occurs at 15 ft.-lb. impact resistance. This is probably an adequate basis for carbon-steel vessels judging by service experience, but more consistent results could be obtained by the use of a fracture appearance test or ductility (percentage lateral contraction or expansion) criterion. This should give a better appraisal of a wider range of steels. A minimum of 10 per cent shear, an average of 20 per cent shear on the fractured surface or 1½ per cent lateral contraction are proposed. Other types of tests could be used and the various methods could probably be made to correlate with this basis. For the purposes of this paper the temperature at which a specimen fractures with a 10 per cent shear failure of 1½ per cent contraction at the notch is defined as the transition temperature T_A. This is approximately the same temperature as T_A on Figure 1.

This determination of T_A is reasonably straightforward when the specific material used is tested. For the immediate future, one must continue to use and accept steels without tests or even special specifications. Particular attention must, therefore, be given to untested steels. This will require some group, such as the ASME Boiler and Pressure Vessel Committee, to establish, with a reasonable degree of accuracy, the expected transition temperatures of available, nonimpact-tested carbon steel. Since there will be an appreciable variation even within a given specification, two more variations of T_A must be defined. T_A (average) is defined as the expected transition temperature for a given steel, but some samples may exceed this value by 20 or 30°. T_A (maximum) is defined as the maxium transition temperature possible for that particular steel.

The various types of tested and untested materials are shown below:

Material test classification	*Description*
A	Rimmed, semikilled or killed carbon steels to standard ASTM specifications without brittle-fracture test requirements and used on the basis of the transition temperature conservatively expected for that specification. This grouping includes steels with and without added requirements for chemical and deoxidation practices to control brittle properties.
B	Brittle-fracture tests required for each heat or heat treatment, but can be made on arbitrary bars rather than the material item itself.
C	Brittle-fracture tests required for individual material items such as each plate, shape, forging or casting. This test must be made after heat treatment is performed. Test specimen must be taken from item itself.

The design considerations proposed are shown in Table I. Basically, all service is limited to the transition temperature T_A. A 20°F margin is permitted for vessels in atmospheric service since the lowest ambient temperature occurs so infrequently that the added risk is not large. In addition, a distinction is indicated between vessels for average and critical services by requiring a greater temperature margin between the design and transition temperatures for vessels in critical service. These services are defined in reference 12, but average service is essentially static or low-cycle loading under relatively nonhazardous conditions, while critical service denotes cyclic and shock loading under hazardous conditions.

Since the transition temperature of actual items in material class B is not known with the same assurance as that of class C, a temperature advantage of 20°F is granted class C as the only practical method to compensate for this uncertainty. This variation between the test specimen location and other locations in material of class C is not considered in deference to current practice and past experience. However, this assumption must be given future consideration.

Service below the transition temperature is permitted only with a safety factor of 20 which should avoid any plastic flow with pressure loads at the

TABLE I—*Design Limitations for Brittle Materials*

Material		Design Nom. S.F.	Limitations					Fabrication		Transition Temp. T_A Req'd.	
					Service						
	Class		Max. Size	Max. t	Max. Press.	Min. Temp. °F	Contents	Hydr. Test Press.	Stress Relieved *	Average Service	Critical Service
Non-Tested Material	A	20	100 cu. ft.	1"	300	**	Non Lethal	7.5 P	Yes	T_A Av'g. $\leq T_o+90$	——
									No	T_A Av'g. $\leq T_o+70$	——
		4	No Limit	**	No Limit	**	"	1.5 P	Yes	T_A Max. $\leq T_o$	——
									No	T_A Max. $\leq T_o-20$	——
Tested Material	B	4	No Limit	No Limit	No Limit	−50 For Plate No Limit On Other	No Limit	1.5 P	Yes	T_A Test $\leq T_o-20$	T_A Test $\leq T_o-50$
									No	T_A Test $\leq T_o-40$	——
	C	4	No Limit	No Limit	No Limit	Mat'l. No Limit	"	1.5 P	Yes	T_A Test $\leq T_o$	T_A Test $\leq T_o-30$
									No	T_A Test $\leq T_o-20$	——

NOTES: P = Design Pressure.
T_o = Lowest operating metal Temp. (°F) except for ambient Temp. Service; For ambient service, T_o=20°F above the lowest expected Atmos. Temp.
* Stress relief would be mandatory when necessary for mechanical properties.
** Determined by predicted transition temps. which will be a function of thickness.

operating temperature. However, since knowledge of a material's limitation is even more lacking below the transition temperature than above, limitations in size, thickness and service conditions are added in order to restrict this usage, while not preventing it for the cases where it is economically attractive. A test pressure of 7.5 P is prescribed since the test may be made at or above T_A and the vessel is adequately strong for such a test. Possibly, stress relief should be required for these vessels but this should be resolved by further investigations.

Stress relief is not indicated as mandatory as a function of service temperatures. However, the generally beneficial results of stress relieving in minimizing the potential of brittle fracture through (1) overcoming partly or completely the harmful effects of cold work and strain aging and (2) reducing residual stresses at welds are qualitatively recognized by allowing a 20°F service-temperature advantage for stress-relieved construction.

When testing vessels where the atmospheric or test fluid temperature is lower than the transition temperature T_A for which the vessel is designed, the test fluid or the vessel shall be heated so that the minimum metal temperature is above the transition temperature. In addition, for large elevated or other field-fabricated tanks tested in positions where the release of the liquid content would be very hazardous, the test temperature of the vessel material and contents shall be above the temperature T_A plus 30°F. Similarly, vessels tested with air or gases shall have the gas and metal temperature above the maximum or design T_A plus 30°F due to the high energy content involved. Unless the design is based on a safety factor of 20, no maintenance or pressure tests shall be applied to vessels when the metal temperature is lower than the average T_A temperature. Pressures applied to vessels designed on a safety factor of 20 shall not exceed the stamped design pressure at temperature below the average T_A temperature.

EXTREMELY LOW TEMPERATURES

The problems of extremely low temperatures are primarily an extension and application of the philosophy proposed in the preceding chapter. In some respects the brittle problem is less because it is economically more practical to use costly materials such as austenitic steels for extremely low temperatures.

Using the 15 ft.-lb. Charpy keyhole test as a criterion, ferritic materials are available down to roughly—150°F. Special types of carbon steels can be used to −50°F. The 2½ to 3½ per cent nickel steels show adequate impact properties down to −150°F, and 8½ per cent nickel can be used below −150°F. Generally, there seem to be procurement problems with the nickel steels unless a full heat is required. Consequently, a great deal of fabrication in this range utilizes 18-8 Cr-Ni steels. If nickel steel is used, stress relief is required and this is difficult on field-fabricated vessels, though by no means impossible to perform.

The austenitic materials, though they avoid the brittle-fracture problem, introduce other difficulties in welding and unexpected corrosion. To avoid hot

cracking in 18-8 weld deposits, it is common practice to employ a duplex weld structure providing some ferrite. For low-temperature service, the ferrite content must be kept to a minimum to insure good ductility and impact strength. The impact strength of this type of duplex weld decreases gradually from room temperature to −300°F and requires careful investigation of each heat of electrodes.

There are various "novel" types of fabrication which may well be more economic than austenitic or even "special" carbon-steel fabrication. Essentially there are three areas which offer real promise: (1) the use of a shell material which includes a "crack stopper," (2) the use of internal insulation to keep the metal temperature above its transition temperature and (3) fabrication from various plastics which are not subject to brittle failure. These are mentioned only as possible future avenues. The use of "crack stopping" materials stems from the speculation about materials such as wrought iron which contain laminations of scale and iron. It is thought that this type of material would inherently prevent cracks from running in much the same way that a brittle crack usually stops at a riveted joint. There are instances, however, where a brittle crack has run through a riveted joint. If this thesis is extended to vessel construction then the laminated vessel seems to show considerable promise. With investigation as to the chance of a crack running from layer to layer it might be conceived that the probability of more than a few layers failing abruptly is low enough so that statistically the vessel is reasonably safe. Some of the difficulties with this type of construction are the end closures and nozzles which cannot be laminated. Any heads used in vessels of this type must be laminated or of material with adequate resistance to brittle failure. The end forging must be made of a steel with a transition temperature meeting the requirements of Table I. Another avenue which bears further investigation is the entire field of glass-filament-wound and plastic pipe.

THE CHALLENGE OF THE FUTURE

The uncertainties in evolving a truly effective design basis to prevent brittle fracture are the real challenges which must be solved in the future. At present, it is necessary to operate in a range between temperature T_B, where one is sure that any failure will be ductile and not brittle, and temperature T_A, where the probability of a failure being brittle is extremely high but where the probability of initiating a disastrous failure is not established but appears to be low.

There are two avenues which must be pursued. One is in the area of a clearer understanding of the actual physics of brittle failure. The SOD tests must be made to check quantitatively with the Orowan extension of the Griffith crack theory. Also, enormous improvements in the metallurgy and manufacturing of carbon steels with good brittle properties are possible. With the proper combination, brittle-resistant carbon steels should cost the same as carbon steels currently in general use. It is important to strive for an accepted understanding of the brittle-fracture problem by both the user and the producer of steels.

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MARINE ENGINEERING NOTES FROM THE SOVIET PRESS

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SIXTH FIVE-YEAR PLAN ENDED

On 1 January 1956 usual Soviet fanfare ushered in what will go down in the books as the Sixth Five-Year Plan, a plan which, for various reasons, was suddenly terminated on 31 December 1958, in favor of a new, seven-year plan. The reasons for the cancellation are many and varied, subject to much speculation as to basic causes which led to the rather surprising action on the part of the Soviet government. Previous plans have been completed ahead of time, one was cut short by the outbreak of World War II, others have run their full courses, but the Sixth is unique in that Soviet speech makers claim the plan was curtailed because it was not broad enough in scope.

Final figures are not yet available for the total run of the Sixth prior to cancellation, and, in any case, would require considerable analysis before a relatively firm basis could be arrived at for a decision as to the "why" of the action. Yet, without going too deeply into causes, one may suggest, for the shipbuilding industry at least, that there has been an underlying inability to utilize fully the capabilities available, to marshal these forces and direct them to fulfillment as ordered. There is little doubt that the distribution of shipbuilding and ship repair yards among many ministries and organizations has made it difficult for Soviet planners to bring about the specialization and cooperation needed among the enterprises and shipyards.

The Ministry for the Maritime Fleet, the Ministry for the River Fleet and the Ministry for the Timber Industry are all engaged in shipbuilding and ship repair, while, at the same time, the former Ministry for the Shipbuilding Industry (now the State Committee for Shipbuilding) and others, in addition to their activities in shipbuilding and ship repair, are engaged in producing iron, steel, rolled products. In fact two steel mills are subordinate to such organizations. The machine building plants subordinate to these groupings operate at a loss because shipbuilding and ship repair orders have totalled a meager 8 per cent of the total capacity of the plants. Other plants under the old Ministry for the Shipbuilding Industry included those engaged in the manufacture of radio receivers and transmitters, telephones and other equipment manufactured on a large scale by the plants of the radio industry. At the same time that its own plants were operating below capacity,

the Ministry was placing orders for its ships with the radio industry. Instrument building plants came into being which, in addition to shipbuilding orders, began to fill orders for general industrial use. For example, plants manufacturing lighting equipment for shipboard use have also produced for industries subordinate to four other ministries.

Enterprises affiliated with shipbuilding manufacture small diesel engines, steam pumps, turbo-generators, compressors and other equipment, despite the fact that specialized ministries manufacture the same products. At the same time the design and building of small craft is indulged in by various ministries, organizations, individual enterprises, various volunteer societies and the like. The design bureaus of the water ministries, as well as the timber industry and other organizations have developed various types of cutters and small craft for particular needs. Since the various types are designed for special purposes there is little standardization and costs, as a result, are high. In 1956, for example, ten shipyards subordinate to the Ministry for the River Fleet built about 150 river passenger cutters of five different types and sizes in the 150-horsepower class, as well as approximately 200 dry cargo and tanker barges of various types and sizes in the 150-ton capacity class. In that same year, 150-horsepower tugs were built in the shipyards of five different ministries. It is suggested that it would be possible to build small craft cheaply if the building programs were channeled into specialized shipyards and undertaken on a large scale on the basis of a standard design. Costs are high at present because shortages of ordinary cutting tools, fittings, machinery spare parts, electrodes and other items exist in the shipbuilding industry. It is said that cutting tools are made by hand at a cost of from four to six times that of tools produced in special factories.

Certain of the shipyards, for reasons which include better administration within the over-all framework of administrative checks and balances, have provided data which restate the simple economic fact of lower unit costs when production is undertaken in quantity. In one shipyard it was determined that the cost of the 20th ship of a given type was approximately 45 per cent lower than the cost of the first ship, and at the same time that the man-hours required for construction for the 20th ship were 35 to 40 per cent lower. Another shipyard, building 3,700-ton capacity barges, presented the following figures for its production cycle:

Ship in series	Year of delivery	Manhours in percent of lead ship	Cost in percent of lead ship	Costs for materials and prefabricated parts in percent of lead ship	Costs for wages in percent of lead ship
1	1950	100.0	100.0	100.0	100.0
3	1951	87.0	72.5	90.5	89.0
6	1953	68.8	61.3	73.9	69.6*
10	1954	59.0	58.8	61.4	73.4
11	1954	56.1	54.3	58.2	70.0

* As stated. A possible wage increase may account for the peculiar sequence or a possible typographical error.

Finally, a shipyard engaged in the serial production of 3,000-ton capacity steel barges reported that the cost of the 120th unit (the program at the time was in its fifth year) had been reduced by more than half the original cost, and the manpower required to build the unit was 60 per cent that of the original. The yard was, at the same time, building 200-horsepower tugs and two types of suction dredges. The barge program was, however, undertaken in a separate section of the yard and advanced construction methods were in use.

Investigation of these, and similar figures obtained from the *Kazbek*-class tanker program (inaugurated about 1949, with *Kazbek* in service in 1951) led a Soviet author, I. I. Uzor, to say in a 1956 book titled "Overhead Expenses and Ways to Reduce Them in Socialist Industry," that "as a result of the shift to building identical ships the reduction in the cost of construction for self-propelled ships has been reduced 30-35 per cent, while the cost for non self-propelled ships has dropped 35-40 per cent and, at the same time, overhead expenses have been halved as a result of the improvement in the organization of production."

Nevertheless there were serious faults in the inter-plant cooperation program for 1956 (the year, apparently, when, despite the beginning of the Sixth plan, doubts began to arise as to its ultimate fate) within the shipbuilding complex. The plan required that boats built in Leningrad be sent to yards in the Far East, at the same time that the identical boat was being built in Vladivostok; boats built in Sosnovka in Kirov Oblast were, in turn, sent to Leningrad. One of the southern yards was ordered to send steel castings to Leningrad and to the Far East, while the Odessa machine building plants were to receive steel castings from other cities. Leningrad has been cited as a shipper of over 100,000 tons of iron and steel castings annually, at the same time that it receives as much as 40,000 tons of castings from other cities.

As a result of decisions taken at the XXth Congress of the Communist Party certain changes have been made; the transfer of diesel engine building to the transport machine building enterprises; the establishment of a Ministry for Instrument Building and Automation; the organization of centralized electric welding electrode production for general use within the system of plants under the ferrous metals industry of the Soviet Union, while elec-

trodes for special uses will be manufactured in plants in other branches of industry.

Within the same time span there came about the industrial reorganization which subordinated many of the entities, including shipyards, to the Councils for the National Economy (SOVNARHKHOZ). Apparently aimed at decentralization, there is some evidence that the reorganization has substituted one type of bureaucracy for another. In Moscow some thirty instrument building plants were consolidated under the control of the city SOVNARKHOZ which, in order to exercise supervision, had to establish what is known as the Administration for Specialization and Cooperation. Since this type of organization concerns itself with future development, it will, in itself, become the colossus it was meant to replace.

UNDERWATER LABORATORY IN SERVICE

The submarine converted to "scientific" purposes has been named *Severyanka* and, late in 1958, completed a ten day voyage during which scientists and correspondents were treated to a view of underwater life they had not previously seen.

A description of the voyage, by one of the correspondents, suggests that, despite the difficulties anticipated by Soviet engineers (See the JOURNAL, November 1958, p. 619) port holes have actually been installed in the hull and the submarine has been equipped with both "short-range" and "long-range" searchlights, the latter illuminating the depths to a distance of 60 to 65 feet.

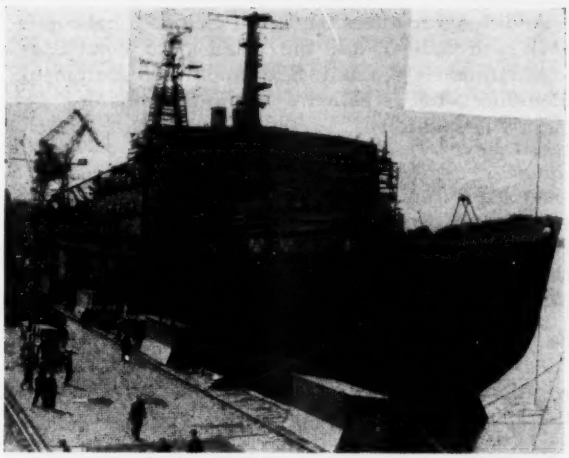
The initial experiment in the use of a craft of this type was carried out in the Barents Sea under the auspices of the All-Union Scientific-Research Institute for Maritime Fishing and Oceanography (VNIRO) and included scientists from the Polar Institute for Maritime Fishing and Oceanography (PINRO). The expedition was led by VNIRO's Director, V. P. Zaytsyev.

The equipment which seemed to attract most of the attention was a fathometer mounted so as to probe the water depths *above* the submarine. The boat submerged for several hours each day and was obviously, although not specifically reported to be, under a Soviet Navy crew.

Upon completion of the trial voyage the *Severyanka* returned to an undisclosed port, believed to have been Murmansk on the basis of a later report from that city that *Severyanka* had, on 1 January departed on her second voyage. This voyage was to have lasted for 25 days and was announced as the first of "several" to be made during the course of 1959. While the area of the research was not announced for the second voyage, it had earlier been noted that later trips would be out into the open ocean—suggesting probing voyages into the Arctic and north Atlantic oceans.

ATOMIC ICEBREAKER *Lenin*

It is now established that *Lenin*, laid down in the Admiralty Shipyard in Leningrad on 25 August 1956, and launched on 5 December 1957, was well



The atomic powered icebreaker *Lenin* pictured late in 1958, scheduled for trials in the spring of 1959, is nearing completion.

advanced at the beginning of 1959 (see cut) and was, in fact, scheduled to hold dock trials in April. Certain trials were reported as having taken place late in 1958 and early in 1959 for, if schedules were met, all three reactors, as well as propulsion and other machinery, had already been installed in the vessel.

Each of *Lenin's* reactors is said to be three and one-half times larger than the first reactor used by the Academy of Sciences of the USSR to produce electric power on a commercial scale. It may be recalled that the first reactor, developing a useful power of 5,000-kilowatts, was put in operation on 27 June 1954. *Lenin's* reactors are said to be pressurized water units, as compared with the graphite-moderated water cooled unit first developed for commercial use. Each of the ship's reactors has an independent primary circuit carrying cooling water to a separate steam generator where superheated steam is made in the secondary, water loop. The steam is then led to the four main turbo-generators, each of which is rated at 11,000-horsepower, to auxiliary turbo-generators and other steam auxiliaries. Available data on characteristics include:

Length overall, feet	440.0
Beam, greatest, feet	90.6
Horsepower	44,000
Speed, normal, knots	18
in 7.5 feet thick ice, knots	2

Lenin is fitted with three propellers. The weight of the atomic energy installation is given as 2,970 tons, including shielding. Comparative data for *Lenin*, *Savannah* and the T-5 tanker with atomic energy installation follow:

	Ratio Displacement/Horsepower	Efficiency Shaft Horsepower/Reactor Power
Lenin	1:2.75	.37
Savannah	1:0.92	.24
T-5 Tanker	1:0.70	.27

Additional features for *Lenin* include a helicopter deck and hangar aft, the usual gyro compasses, logs, fathometers, radar, RDF and other equipment, including what is called a "radio-coordinator," not otherwise identified.

TANKER CONSTRUCTION PROGRAM

At least four classes of tankers are under construction at the present time in Soviet shipyards. The types range from the "super" tanker down to river tankers of various capacities and appear to be carry-overs from the Sixth Five-Year Plan program.

Super Tanker

About two years ago plans for a "super" tanker were announced. The vessel was laid down in the Baltic Shipyard in Leningrad and was launched on 4 December 1958. Characteristics for what Soviet builders say is "the largest ship ever built" in the Soviet Union include:

Length, feet	665.4
Beam, maximum, feet	84.6
Height of Side, feet	44.9
Draft, feet	34.9
Capacity of 30 tanks, tons	26,500
Horsepower	19,000
Speed, knots	18
Cruising radius, miles nautical	10,000
Displacement, tons	40,000
Propeller, five bladed, 30 tons, 22.3 feet diameter	

Named *Pekin*, the tanker is of all-welded construction and is scheduled to carry three types of products simultaneously between the Black Sea and the Far East. The scope of this program is unknown beyond the fact that a second tanker was reported laid down in the same shipyard late in December 1958. As an indication of the manner in which inter-plant cooperation works in the industry, it has been noted that the steam anchor windlass for *Pekin* was designed and built in the "Leninskaya Kuznitsa" Shipyard in Kiev. The windlass is said to be powerful enough to lift two anchors, each weighing 15,400 pounds, simultaneously.

Kazbek-class

Soviet shipyards have, since the completion of the class leader in 1951, added fifty others of the class to the Soviet tanker fleet. However, from a peak delivery of twelve in each of the years 1956 and 1957, deliveries fell to but four identified as of the class in 1958. No reasons for the curtailment of the program have been given but it may be speculated that the super tanker class has been substituted for the oft-maligned *Kazbeks*. Two of the four delivered were built in Kherson and one each came out of Nosenko and the Admiralty shipyards.

Shallow Draft

The first of the class was *Oleg Koshevoy*, delivered in 1955. (See the JOURNAL for May 1957, p. 312.) Seven of the class were built on the same design in what has been called the "first series." A "second series" began with *Inzhener A. Pustoshkin* in 1957 and this series now numbers seven as well. Four deliveries occurred in 1958, the precise number delivered in 1956 and 1957 and appears to be a stable delivery schedule for the one shipyard which is building the class at the present time. In all, fourteen *Oleg Koshevoy*-class tankers are presently in operation on the Caspian.

River Tankers

Supplementing the information previously made available on the 3,000-ton capacity river tanker building in the Stalingrad Shipyard on the Volga River is the following:

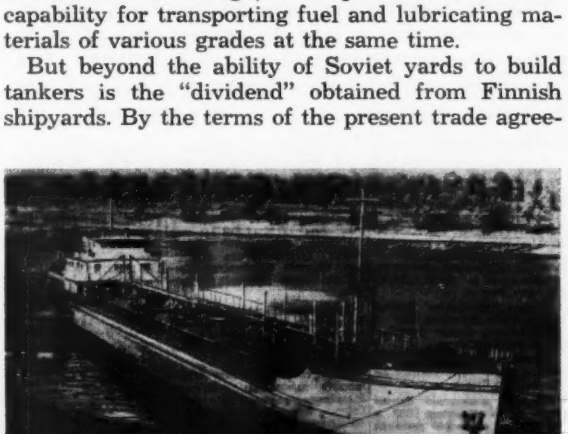
Length overall, feet	360.9
Length between perpendiculars, feet	351.1
Beam, feet	42.7
Height of side, feet	15.7
Draft, feet	11.5
Capacity at above draft, tons	
Gasoline (SG 0.74) in all tanks	2950
Viscous products (SG 0.92) in centerline tanks	3250
Main engines, Bakau-Wolf R5DV-148	2 x 500
Speed, designed, at 345 rpm, knots	9.0

The dimensions will permit the tanker to operate not only in the Volga-Kama basin, but along the Volga-Baltic water route and through the Volga-Don Canal as well.

Yet another river tanker, this in the 150-ton capacity class for use on the shallower rivers of the Soviet Union, was to have been launched late in 1958 by the Lenin Shipyard in Astrakhan. No further details, beyond the fact that the series would continue into 1959, were made available.

Finally, a 600-ton capacity tanker for use on the reservoirs and lakes, as well as on the rivers, was accepted by the State Commission in mid-December. This series is building in Ryibinsk (Shcherbakov) in the Volodarskiy Shipyard. Incorporating five tanks in the design, the ship is said to have the capability for transporting fuel and lubricating materials of various grades at the same time.

But beyond the ability of Soviet yards to build tankers is the "dividend" obtained from Finnish shipyards. By the terms of the present trade agree-



First of the 3,300-ton displacement river tankers shown testing machinery in the Stalingrad Shipyard.

ment between the two nations, Finland, during the period 1956-1960 is required to build and deliver twelve 3,300-GRT tankers at the rate of two per year, except for 1958 and 1960, when the contract is for three. Another class, of 1,100-tons capacity, concluded in 1958 with the delivery of the last of eight programmed.

According to the Minister for the Maritime Fleet of the USSR, the Finnish tankers in the larger series are all destined for service in Far Eastern waters where there is an obvious need for relatively small tankers capable of transporting petroleum products to the various shallow-water points in the basin. 1958 deliveries may have included *Vilzhyisk, Mozyir* and *Kokand,* launched in Rauma, Finland, between December 1957 and July 1958.

FINNISH-BUILT ICEBREAKERS FOR USSR

According to the terms of the 1956-1960 trade agreement between Finland and the Soviet Union, the latter was to receive delivery of a 20,000-horsepower icebreaker in each of the years 1958 and 1960. However, the usually prompt Finns will be late in delivering the first icebreaker because *Moskva,* as the new vessel is named, was not launched until 10 January 1959.

The 13,900-ton displacement icebreaker, largest of its type to be built in Finland, will be rated at 22,000-horsepower and will measure:

Length, feet	406.8
Beam, feet	85.3
Height of side, feet	34.5

The main propulsion machinery will consist of eight diesel engines coupled to individual generators developing 2,200-kw each. These generators will provide power for the four main drive motors which, in turn, will enable the ship to break ice up to 20 feet in thickness, each motor rated at 5,500-horsepower.

The West Berlin firm of Siemens-Schuckert is supplying the building yard, "Khiyetalakhti" (Wartsila) in Helsinki, with the main propulsion machinery as well as with the majority of the electrical equipment with which the two will be equipped.

Present Soviet plans are said to call for the two units to operate in the eastern sector of the Northern Sea Route.

PLASTIC BOATS

Engineers attached to the Central Technical-Design Bureau of the Ministry for the River Fleet of the RSFSR, under the supervision of Chief Designer M. G. Avrukha, have designed a plastic cargo motorship 54 feet long able to carry 15-tons of cargo. It has a folding hatch cover and is equipped with a two-place cabin in the bow. Engine and rudder controls are centralized.

Under construction in one of the river yards, the plastic cargo ship is assembled from three basic sections: hull, decks complete with wheelhouse, and forecastle. Preparation is said to be simple. A wooden form is covered with an insulating wax emulsion which is coated with successive layers of polyethylene resin and glass wool. The resin, penetrating the wool, combines with it and hardens in the course of 15-60 minutes. When removed from the form the ship section needs no further finishing since the surface is smooth. The plastic composition used is as good, strength-wise, as the better grades of steel, yet is only one-fourth as heavy. The individual sections are joined together with the use of a special glue.

The cargo motorship is the third type of plastic craft designed by the Bureau. Previous efforts include the design, and construction, of life boats and motor cutters in the 12-14 man size.

The cutter, weighing 2,950 pounds, is 25 feet long, 6 feet in beam and draws 1.1 feet. Powered, it will make 15 knots. A cutter of this type was tested on the Khimki Reservoir of the Moscow Canal in the fall of 1958. This, together with the larger vessel noted above, is a product of the imeni Kalinin Shipyard, where work was done in collaboration with the Plastics Institute, as well as with the Design Bureau.

WINTER SHIP REPAIR PROGRAM

In the Black Sea, Northern and Caspian, as well as in other steamship companies, the winter of 1958-59 was one in which, in addition to usual repair work, the Soviet planners laid on a large modernization program. This program, as planned, included the conversion of coal burners to liquid fuel users, the installation on cargo carriers of mechanically operated hatch covers, or mechanization of existing covers to facilitate opening and closing, the introduction of automatic combustion control systems for boilers, as well as other items.

River ships were included in the modernization program as well, with major emphasis, as in past years, on conversion of both tugs and barges to facilitate the use of the push method of towing, the reinforcement of such vessels to permit their use on the open reservoirs and on the artificial seas, the installation of remote engine controls on the *Bolshaya Volga*-class (over 200 of which are presently in service throughout the Soviet Union) so as to permit personnel on the bridge to handle engine changes directly, the widening and lengthening of hatch openings to more readily facilitate loading and unloading of steel barges, and the installation of heating systems in petroleum carriers to permit greater ease in pumping cargoes and to facilitate cleaning the tanks.

Of these items, that of the conversion of steamships to burn liquid fuel is a continuing one. All river steamers are scheduled for conversion to oil burners by 1962, with the exception of those units operating in areas where such conversion would not be economically feasible. As of December 1958, however, over 550 had already been converted and from this change alone Soviet sources claim the re-

lease of 1,500 firemen to other assignments. Other advantages which have accrued are said to include a reduction in wages and fuel funds amounting to several million rubles, a reduction in the time spent in bunkering with a corresponding increase in productivity, and a lightening of the work load for the engine room force. The recently concluded repair period called for the conversion of an additional 200 Volga, Irtyish and other coal burners to oil firing. Initial progress reports suggested that the program was lagging and efforts were made to speed up the work by pointing out certain advantages to be obtained from successful accomplishment.

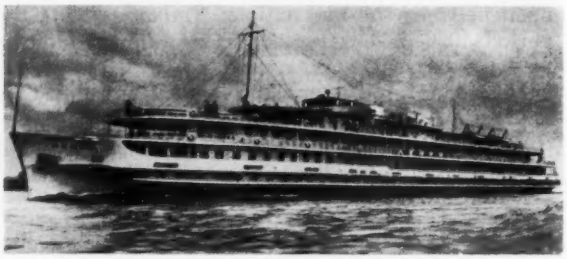
For example, if the 70 river ships scheduled for conversion to use remote engine controls were actually completed in the 1958-59 winter period, the number of crew members actually assigned to these ships for the current season has been halved, with a savings in wages of 1,700,000 rubles anticipated. The present day version of the *Bolshaya Volga*-class is known as the *Shestaya Pyatiletka*, the crew of which was reduced from 19-21 to 12 as a result of the remote engine control installation, and serves as a specific example of the reductions which are said to be taking place. However, two of the organizations engaged in the manufacture of the remote control systems, one of which was the Gorodets Marine Machinery Factory, were not supplied with all the technical documents and equipment needed to manufacture the parts, suggesting something less than complete fulfillment of the plan.

The barges selected for hatch enlargement included 50 of Project 425 (believed to be a 2,000 ton capacity steel barge) but at the end of 1958 only 5 of 20 planned were under conversion in the Astrakhan shipyards, and none of the 25 scheduled by the Volga Steamship Line had been started. The Volga Tanker Steamship Line was scheduled to fit heating coils in 42 of its barges by this spring in a program dating back to last year. It is indicated that only 19 were so equipped in 1958, so that the past repair period work load was 23 barges, only one of which was under conversion at year's end. One reason for the delay in starting work was the fact that much of the tonnage had not been cleaned up when the season's work ended and yards will not accept such dirty tonnage.

Finally, the White Sea-Onega Steamship Line undertook a program for the strengthening of a group of lighters in the 1,000 and 2,000 ton capacity classes for service on the open sea as part of an expansion program which has seen this particular line expand its operations beyond the river mouth onto the northern seas.

NEW COAL-ORE CARRIER

One of the major cargo runs in the Black Sea area is that connecting Poti-Zhdanov-Odessa, a run over which coal and ore are moved by two basic classes of cargo carriers. The two classes, *Pervomaysk* and *Chulyim*, products of Polish shipyards,



The new passenger diesel-electric *Lenin*.

capable of lifting 2250 and 2600 tons, respectively, have been a source of considerable trouble to the Black Sea Steamship Company. Some twenty *Perovmaysk's* are in use on the Black Sea, while the number of *Chulyim's* is perhaps twenty-five.

Both classes have required extensive alterations and have proven to be uneconomical in operation. These facts, together with the new-found desire of Soviet planners to emulate Western operators in turning a profit from their endeavors have resulted in plans for a new diesel-propelled coal-ore carrier. Some of the characteristics are:

Length overall, feet	457.7
Length between perpendiculars, feet	426.5
Beam, feet	59.0
Height of side, feet	33.8
Draft, loaded with ore, feet	26.2
with coal, feet	25.7
Draft, in ballast	
forward, feet	11.1
aft, feet	18.4
Capacity, ore, tons	8,980
coal, tons	8,632
Hold capacity, cubic feet	424,000
Cruising radius, miles	6,000
Displacement, empty, tons	4,200
Speed, knots	14.3
Crew	37

The vessel's dimensions were selected by designers from the Central Planning-Design Bureau No. 2 in Moscow, and approved by the Ministry for the Maritime Fleet as meeting requirements for use in Kerch Channel and in the shallow Sea of Azov. However, the vessel will also be able to meet unlimited sailing requirements, permitting its use on the high seas if such use is required.

The engine is a product of the East German "Dizelmotorenverk" in Rostock and is designated K7Z-70/120C. It is described as a two-cycle, 7-cylinder, crosshead, single-acting engine, turbo supercharged, rated at 5,400-ehp at 115-rpm. Normal fuel consumption is 155 grams/ehp/hour ±5 per cent. Two turbo compressors provide for supercharging. A closed system of fresh water cooling is used for engine and injectors and the total weight of the engine is 333 tons.

The main ship's service generators are diesel driven by 60DV136 engines rated at 300-ehp at 500-rpm. These engines are 4-cycle, 6-cylinder and drive

the three alternating current generators. Air starting is provided by a 12-hp diesel driving a charging compressor.

The engine room, located between frames 12 and 46, also contains auxiliary machinery, tanks and piping. A machine shop will be located on the platform and the auxiliary steam boiler is located on the main deck.

The steering engine is electric-hydraulic, capable of providing rudder drive from full-to-full in 30 seconds. The rudder itself is welded, balanced, of the Simplex design. The anchor engine is electrically driven, rated at 9.2 tons and will bring up an anchor at the rate of 59 feet per minute. Three anchors are fitted, one a spare, of the Hull design, weighing 8,800 pounds each. Chain is cast steel 1.4 in. link, and a total of 900 feet of chain is provided. An Admiralty-type stop anchor and a warping anchor, 2750 and 1,100 pounds, respectively, will also be fitted.

Cargo handling equipment is to consist of five electric cranes of 5-tons capacity each; longest boom reach is 16 feet. Hatch covers will be mechanical, on the MacGregor design, covering openings measuring 39 x 34.5 feet.

Two 49-man capacity life boats, fitted with a newly developed hand drive, will be taken by electrically operated davits designed to permit launching when the vessel is listing 15 degrees.

SHIPYARD NOTES

Kalach

This Don River shipyard has built a 150-hp waterjet propelled tug, the largest of the type yet built in the Soviet Union. With a draft of not more than 31 inches, the tug will be able to navigate on the very shallow rivers. Rated at 10 knots, it is of sufficient power to tow barges of up to 300-tons capacity.

Khabarovsk

A Far East shipyard, Khabarovsk, late in 1958 delivered a fishing trawler for the Pacific Ocean fishermen. Built on plans furnished by the "Leninskaya Kuznitsa" Shipyard in Kiev, this was the lead trawler of a new series to be built by the yard.

Kherson

The Kherson Shipyard built a four-section, reinforced concrete floating dock for the Zavkavkazskaya Federatsiya Shipyard, Baku, which necessitated towing of the individual sections to destination via the Volga-Don Canal. When installed in its special berth in the shipyard the dock, 426.5 feet long and 111.5 feet wide, will be able to take the largest ships on the Caspian Sea, its lifting capacity being stated as 6,000-tons. The individual sections were buoyed by pontoons filled with air in order to decrease the draft of the tow, which got underway in June 1958. By October three of the sections had arrived in Baku, with the last section expected shortly thereafter.

The dock is said to be the first of its size and type on the Caspian and is equipped with its own machine shop and storerooms, as well as with crew compartments and living spaces. One man will be able to flood and dewater the dock from a central control station and an intercom system will enable the dockmaster to communicate from any one of six transmitter stations installed at various points.

Kherson was also assigned the construction task of building the first of a series of geared turbine cargo ships, Leninskiy *Komsomol*. Progress may be measured by the fact that late in 1958 a Captain Baglay was assigned to the vessel as master. Characteristics presently available include:

Length overall, feet	557.4
Beam, feet	71.5
Height of side, feet	42.3
Capacity, maximum, tons	13,400
Horsepower, effective	13,000
Speed, knots	18
Crew	42

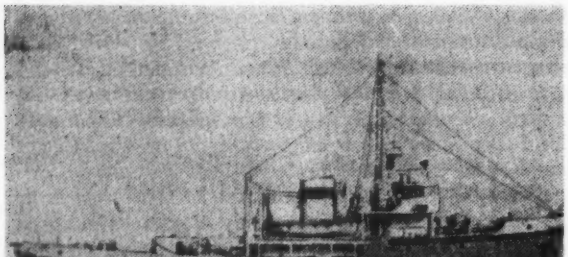
The vessel is a two-decker, full scantling type for carrying dry, general and/or grain cargoes in six holds. Ice strengthened, the vessel will have a cruising radius of 12,000 miles, propelled by a geared turbine unit operating on superheated steam provided by two boilers, each of which will produce 25 tons of steam per hour. Electrically driven auxiliaries will obtain power from two turbogenerators rated at 800-kw each.

A second vessel in the class has been assigned to the I. I. Nosenko Shipyard, also in the Black Sea area. However, there is little likelihood that production has reached the stage where vessels of this type will be for export as was indicated when a model of the vessel was displayed at the Leipzig Fair last spring.

Klaypeda

The Baltic Shipyard in Klaypeda produced its first cargo motorship, named *40 LET VLKSM*, for the Western Steamship Line, a river organization. The vessel will be used to carry cargo over what has been named the "Friendship Line," a run connecting river ports in the Soviet Union with ports in East Germany; Kaliningrad-Magdeburg via the Wisla and Oder rivers. Designated Project 898, plans call for a 300-ton capacity vessel 166.7 feet long, 23 feet in beam, drawing 3.9 feet empty and 4.5 feet loaded (stern measurements). The 150-horsepower engine will drive the vessel at a speed of 7.5 knots when loaded. The class is also building in the **Kaliningrad Shipyard**, the first from this latter yard having been named *Ukraina*.

Chief Designer V. Gapanovich, of the Central Technical-Design Bureau of the Ministry for the River Fleet, developed the vessel to cope with the problems of locks and bridges in East Germany and, as a result, the maximum height of the vessel when empty is only 12 feet and the upper portion of the wheel-house, as well as the mast, can be dismantled by folding down.



Seagoing Tug *40 Let Vlksm* built in the Vano Sturua Shipyard in Baku on the Caspian Sea.

imeni Komintern

Scheduled to build ten 150-horsepower passenger motorships in 1958, the Kherson located yard is known to have completed work on three of them by the fall of 1958. One, *Povazhnyiy,* was turned over to the Lenin section of the port of Zaporozhye on the Dnepr River. In addition several tanker barges and a self-propelled ferryboat were launched.

Komsomolsk

Little known, and mentioned infrequently, this shipyard on the Amur River in the Far East may well be an important builder of various classes of merchant and naval vessels. In recent years the yard has delivered two passenger ships to the Far Eastern Steamship Company. Named *Zabaykalye* (1956) and *Priamurye* (1958). the vessels are diesel-electrics, capable of carrying 600-700 passengers. Both are, or have been, operating between Vladivostok and Korsakov.

Krasnoyarsk

Complaints are beginning to filter in about the quality of the water-jet propelled cutters built by the yard. It is said that the living spaces are poorly finished, that low grade veneer has been used throughout. Leaks were found in all the cabins after the first rain was encountered; doors, windows and the rubbing strakes are all ill-fitted.

Krasnoye Sormovo

The builders of the hydrofoil *Raketa* are reported building a series of a smaller version of the original. The small version is a six-place runabout said to have made 36 knots in trials on the Volga River last summer. At the same time, an improved version of *Raketa* is also under construction. The new model is said to be more maneuverable and better finished. It is rated at 800-horsepower and can carry 60 passengers, a few less than *Raketa.*

The official underway trials of the latest of the many types of river passenger ships was held between 29 July and 5 August 1958 on the stretch of the Volga River between Gorkiy and Gorodets and on the Gorkiy Reservoir. A product of "Krasnoye Sormovo," *Lenin,* as the new three-decked diesel-electric built in accordance with Project 20 KB for the Ministry for the River Fleet. The largest of its type, the 2250/2700-ehp vessel has the following characteristics:

Length overall, feet	398.3
Length between perpendiculars, feet	380.6
Beam, greatest, feet	55.1
Height of side, feet	50.0
Greatest draft, feet	7.9
Displacement, loaded, tons	2,235

Three type D-50 locomotive type diesel engines have been used for generator prime movers, each rated at 900-ehp. The generators, in turn, feed three main drive motors, each coupled to its own shaft and rated at 750-ehp at 350-rpm.

Design has been such as to obtain for the vessel an "M" rating, permitting it to operate on any of the lakes or reservoirs in the USSR without limitation and in any kind of weather.

Cabins are provided for 439 persons as passengers and the vessel carries a crew of 97. From previous comments it would appear that accommodations for the crew were overlooked, for it is now noted that some of the cabins previously designated for passengers have had to be turned over for crew and other uses.

A second vessel of the class, *Sovetskiy Soyuz,* is under construction and is to be delivered for the 1959 season.

Leninskaya Kuznitsa

Reporting a newly designed diesel-electric multi-bucket dredger under construction for river use, the Kiev shipyard noted that, as distinguished from previously built dredgers, this one will be able to dredge at a depth of 33 feet. Certain design changes appear to have been made in the manner in which the bucket chain is fitted on the roller bearings, winches will be equipped with magnetic clutches, and an automatically operated utility steam boiler will be installed. However, a later note finds a complaint lodged against a new dredge designated as *Yeniseyskiy-6* in that the work of the dredge operators is made more difficult than comparable work on other types of dredgers. Thus, although the operators are allotted 50 to 60 minutes to bring in all mooring lines, because of arrangements on the deck, the workers need 90 to 120 minutes to perform the operation.

Quality control for welded seams has, in the past, been exercised by the use of radio-active cobalt, but this method, for undisclosed reasons, has proven to be unsatisfactory in this particular shipyard. The yard is now using a "new, radio-active isotope" which is better and which has resulted in the obtaining of better quality control.

The yard is also said to have begun the serial production of what is called a "boiler-separator,"

for use on seagoing tugs. The unit is to be used in conjunction with utility boilers installed on diesel propelled tugs. The exhaust gases from the engines are piped into the unit along with the steam from the boiler and the vapor which falls out is then pumped into the hot water heating system, as well as into the sanitary system. When the engines are secured the boiler is used to generate steam alone.

The yard continues to build cargo motor ships in the 600-ton capacity class, adding ST-601 and *Aksay* to the list of river ships in operation. *Aksay* was fitted with a new type of propeller, larger in size than previously used ones, as well as with a hydraulic steering system which permits faster rudder handling and provides for better maneuverability. On trials the cargo ship made 9 knots loaded, 10 knots empty.

Finally, there is under construction the first of a long series of passenger motorships which will be sent to the Ob and Amur rivers. Built for use on both lakes and rivers, the class is all-welded, 253 feet in length, 34.4 feet in beam and will displace 750-tons fully loaded. Two 400-horsepower engines will enable the twin-screw vessel to make over 10 knots. The design calls for the installation of 308 bunks, 79 of which will be first class in one and two bunk cabins.

Loksa

The Loksa Ship Repair Yard in Tallin has developed from one wooden building some ten years ago into what is now said to be a large, industrial enterprise with modern machinery, equipped with its own heat and power station. In addition to doing most of the repair work on ships assigned to the Estonian Steamship Company, Loksa, in the fall of 1958, completed the assembly of the first seagoing vessel built in the yard—a dry cargo shaland.

M. Gorkiy

The production of *Zelenodolsk*-class tug-pushers appears to be reaching a peak. The number of units in service increased from nine in September to 15 at the end of the 1958 season. The class is named for major cities, not only in China, as previously noted, but in the satellite countries as well.

Moscow

Movement over the Northern Sea Route during the 1958 season included several passenger motorships built by the Moscow Shipyard. The craft were moved over the inland system to Arkhangelsk where they were then staged to Siberian destinations. Two units, of the "lake" type, arrived in Novisibirsk on the Ob River in mid-September while another two arrived in the Lena basin.

I. I. Nosenko

The 43,800-ton displacement whaler, *Sovetskaya Ukraina*, largest merchant ship yet attempted by a Soviet shipyard, was launched in September 1958 and has been designated as Register class L*R 4/1-S (Whaler Base). Planned to either augment, or replace, *Slava*, now on her 13th whaling expedition in the Antarctic, the new vessel presents some interesting data when compared with the latter:

	Sovetskaya Ukraina	*Slava*
Displacement, tons	43,800	28,715
Deadweight, tons	26,700	18,960
Length, between perpendiculars, feet	656.2	490.2
Beam, feet	91.2	71.4
Height of side, feet	62.3	50.5
Draft, feet	34.8	34.5
Coefficient of form	0.74	0.825
Main engines	Diesel	Steam
Horsepower	15,000	4,300
Speed, knots	16	12
Propellers	2	2

When *Slava* departed for her 13th expedition about 1 November 1958, she was accompanied by three additional whale catchers built by the I. I. Nosenko shipyard. Thus far seven of the diesel-electrics appear to have been delivered of a possible total of eleven.

The second of the new class of geared-turbine cargo ships, assigned to Nikolayev's Nosenko Shipyard, has been named *Frederik Zholio-Kyuri* (Frederick Joliet-Curie) for the French scientist, a winner of the Stalin (now Lenin) prize some years ago. (see under **Kherson** for details).

Yet another of the trawlers being built for the fishing fleet, named *Lermontov*, has been reported completed, supplementing the program previously reported as under way in the shipyard.

Perm

A new type of tug-pusher, designated the OT-801, powered by two 400-horsepower diesel engines, has been built by the Perm Shipyad. During trials the lead ship, pushing two loaded 3,000-ton barges, developed 7 knots. Installed equipment includes two 25-kw alternating current generators, a towing winch and a one-ton stern anchor. When originally anounced much to-do was made of propeller shaft driven generators for use when the vessel was underway. However, by October the Chief Engineer of one of the class named *50 Let Profsoyuzov* reported that the tug had serious defects in design. In fact, the design of the generator coupling to the shafting had not proven reliable, a charge levied against the remote engine controls as well. The plans drawn by the Gorkiy Design Bureau were faulty because the location of the main engine control station was not in accordance with the Rules for Technical Safety.

Predivinsk

A Yenisey River shipyard, Predivinsk has been building 600-ton capacity barges for several years now. Many of the barges have been transferred via the Northern Sea Route to the Ob and Irtyish rivers, apparently in annual increments. In the past the yard has also manufactured and pre-fabricated houses for Siberian industrial and construction workers, as well as for the river personnel. One such was described as a "two-apartment, standard house." Annual production at one time was 50 such

units. More recently the yard has been shipping timber and house elements to the imeni Pobezhimova Shipyard in Krasnoyarsk, presumably for further fabrication. Barge building has been extended to include a type designed to carry cattle.

Rechitsa

Located on the Dnepr River to the north of Kiev, the Rechitsa Shipyard was completed in the mid-1930's and was a builder of wooden ships until recently when it was announced that the yard was building 100 and 150-ton capacity steel barges for the Kremenchug Hydro-Electric Station. Work was also underway on the construction of the two-decked barracks ships which are stationed at various places along the river to house workers isolated from town facilities.

Riga

With a considerable background of experience in the construction of sea-going tugs and flat-barges to draw on, the shipyard was, in the late summer of 1958, making preparations to build a series of shallow draft vessels for use as cargo carriers along the Estonian and Latvian coasts. At this time the technical project called for a 350-ton capacity schooner, the construction of which is said to occupy an important place in the yard's production program for the current 7-Year Plan.

This program is in addition to the tug-building program, which, at latest reports was being speeded up to complete tug number 17.

S. M. Kirov

The State Planning Commission for the Fishing Fleet designed what is called a "receiver-transporter," bearing the designator PTS-150 which is assigned to the S. M. Kirov Shipyard in Astrakhan for construction in a "quite large" series. Actually a seagoing refrigerator ship, it will be built of steel, all-welded, with the house and engine aft. Classed as L*R 4/1S by the Maritime Register, it will be powered by one 150-hp engine and will have two chill holds to be used for the transportation of fish. A Freon cooling system will be installed to chill the 2300 cubic feet of hold space while a material known as "ekspanizit" will be used for insulation.

Sretensk

Located at very nearly the extreme limits of navigation on the Shilka River, the Sretensk Shipyard recently sent the first of a series of a newly designed lake-river type of passenger motorship down the river to join the fleet on the Amur River. Designers were sent from the "Leninskaya Kuznitsa" Shipyard in Kiev to assist the local personnel in building the all-welded hull and in the installation of the machinery and equipment sent into the yard. A series is planned for the future.

Taganrog

The series of passenger motorships built to service the holiday makers along the Black Sea coast and the southern coast of the Crimea has been completed so far as the Taganrog Ship Repair Yard is concerned. Present production is concerned with the building of seagoing, self-propelled cranes which can also be used as tugs. Some of the 80-place passenger craft were, at one time, sent over the river and canal system to the Caspian Sea.

Tyumen

Production of the self-dumping timber barges under construction in Tyumen may have hit a snag. Although production called for twenty units in 1958, a late fall comment on the program indicated that only eight of the new barges were actually in operation on the Irtyish River.

Ulan Udye

Located on the Selenga River, which empties into Lake Baykal, deep in the interior of the Soviet Union, the Ulan Udye Ship Repair Yard has been a barge building yard for some years, as well as the builder of 150-hp diesel tugs. Most recent production has included the construction of two refrigertor ships named *Bogatyir* and *Rossiya* for the fishermen of the Buryat-Mongolian Republic.

imeni Ulyanova-Lenina

This Volga River shipyard, several times relocated and under various names, nevertheless has a 75 year history of ship construction and repair work. Presently located on the shore of the Gorkiy Reservoir in the vicinity of the city of Chkalovsk, the yard is considered to be one of the largest in the Ministry for the River Fleet complex. It has been enlarged and recquipped with new buildings to house the various shops which, in turn, are said to be equipped with the most modern machinery. Production today includes the building of tanker barges, lake-type barracks ships, clam-shell dredgers, service cutters for use in servicing the buoyage systems, spoil piping and floats. In the design stage is said to be a self-propelled multi-bucket dredge with a capacity of 17,000 cubic feet per hour, as well as self-propelled shalands to service the dredge.

imeni Vano Sturua

The seagoing tug *40 Let Vlksm* completed acceptance trials on the Caspian Sea in November 1958 and was added to the list of new construction from this Baku shipyard. The 1200-hp tugs under construction, and built, are powered by two 600-hp engines of the 6DR-30/50 class, the fuel system for which appears to be causing troubles. One of the chief engineers assigned to an unnamed tug in the series has reported that of the twelve injectors in his main engines not one met the requirements laid down in the instructions.

Zhdanov

Late in August construction of barracks ships for workers in the Azov and Black Sea regions was started in the ship repair yard in Zhdanov. Listed as 138 feet in length, 29.5 feet in beam, with 29 two-place and two single-place cabins, the vessels will be three decks high, with the mess hall and recreation areas on the upper deck. Delivery of the first of the barracks ships was scheduled for the fourth quarter of 1958.

"THE MARINE ENGINEER AND NAVAL ARCHITECT"

HARDENED AND GROUND MARINE GEARS

ACKNOWLEDGEMENT

This is an abstract of a paper presented to the Association of Japanese Shipowners in Tokyo, as published in the 1958 Annual Steam Number of "The Marine Engineer and Naval Architect." The author is E. H. Abt, Managing Director, Maag Gear-Wheel Co., Ltd., Zurich.

WHENEVER a machine part is subject to a high surface load or friction, it is common engineering practice to make the part wear-resistant by hardening and grinding. Rollers for roller bearings, crankshaft bearing surfaces, gudgeon pins, cams of camshafts, plungers for fuel injection pumps and many other parts of combustion engines, for instance, are today hardened and ground. Strangely enough, until some years ago, many mechanical engineers were of the opinion that this general experience did not apply to gears and out of mere tradition marine reduction gears were made of soft steel having a tensile strength of 50 to 70 kg. per sq. mm. This fact is still reflected in the very low loading or K-factors, which are generally prescribed in inquiries from shipowners.

The tooth surfaces of gears are subject to even higher stresses than most of the machine parts just mentioned, as in addition to the contact pressure the sliding speed varies continuously in direction and intensity. At the pitch circle, the sliding speed drops to zero, which affects the oil film.

There are two principal reasons why hardened and ground gears are not yet common practice for merchant vessels and warships. Firstly, suitable hardening systems for larger gears had to be developed. Secondly, adequate gear grinders had to be developed, by which the unavoidable deformations due to the hardening processes could be removed. Further, gears of air-hardened or oil-hardened steel could only be finished to the required accuracy by this method.

More than 40 years ago, the engineers of the Maag Gear-Wheel Co. Ltd., Zurich, recognized the importance of hardened and ground gears and started to manufacture large gear grinders which allowed highly-loaded gears of small dimensions, long life and highest reliability to be built. Today gears with diameters up to 3.6 m. can be handled.

BASIC CALCULATION

Before describing the development of this gear design, the basic calculations for gear tooth loads will be considered. In practically all cases the deterioration of tooth surfaces whether by scuffing, scoring or pitting, is of prime importance when a standard steel of 50 to 90 kg. per sq. mm. is used; tooth breakage due to too high a bending stress is

of secondary importance. When using higher grade steels for highly loaded gears, bending stresses are more and more important and must be considered as carefully as the surface stresses.

The surface stress is calculated according to the well known Hertzian formula:

$$S=0.417\times\sqrt{P\times E\times\left(\frac{1}{r_1}+\frac{1}{r_2}\right)}\text{ in kg. per sq. mm.}$$

where:

P = tooth load per mm.
E = modulus of elasticity
r_1 = radius of curvature of pinion teeth
r_2 = radius of curvature of wheel teeth

In practice, the gear calculating is performed according to simplified methods, as for instance, to the formula used by Lloyd's Register of Shipping, which reads as follows:

$$P=K\times d\times\frac{R}{(R+1)}$$

where

K = load factor
d = pitch circle diameter of pinion in inches
R = gear ratio

Experience with gears gained during the past years has shown, however, that the shear stresses below the surface layer of the teeth are of prime importance. Exhaustive tests carried out by the American engineers Thomas and Hoersch as well as by the Russian engineer Belajeff have disclosed experimentally the shear stress conditions below the surface layer. As is well known, the stress conditions meshing between two teeth are the same as for two contacting cylindrical discs or cylinders, whereby each cylinder has a radius or curvature of the respective tooth. The tests were carried out with plastic discs or cylinders and the shear stresses determined by photoelastic methods.

Figure 1 shows the nature of stress distribution of two contacting cylindrical surfaces. The three stress components X, Y and Z are all compressive and have their maximum on the line of symmetry in the plane of contact, where the components Y and Z are equal, and the X-component amounts approximately to half their value. Below the plane of contact, all three stress components fall off rapidly, especially the Y-component. Since the material is everywhere subject to three-dimensional stresses, it is necessary to determine the equivalent stress in order that use may be made of the mechanical strength data usually available. Undoubtedly the best method of doing so is to combine the stresses in accordance with the Mises-Hencky criterion of failure, now acknowledged by scientists all over the world. The heavy dotted line shown in the figure is the distribution of the equivalent shear stress responsible for the behavior of the material. At the depth where the equivalent stress is a maximum, over-stressing of the material starts, and explains nicely the formation of pitting. In fact, when running such tests with transparent plastic cylinders, after a certain time a ringshaped area under the surface layer starts to change its color and turns black. This deterioration can even go so far that parts of the surface layer chip off.

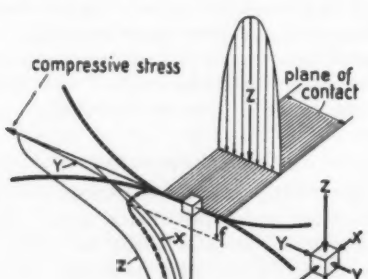


Figure 1. Mises-Hencky equivalent shear stress.

The maximum composed shear stress Sc amounts according to these tests to:

$$Sc\cong 18.6\times\sqrt{P\left(\frac{1}{r_1}+\frac{1}{r_2}\right)}$$

$$\cong 0.3\times S$$

and the depth, at which the maximum is reached:

$$f\cong 0.4\times 10^2\sqrt{\frac{P}{\left(\frac{1}{r_1}+\frac{1}{r_2}\right)}}$$

$$\cong 0.8\times a$$

where "a" represents half the width of the band of contact. These last two formulae are today of the greatest importance in calculating gears.

The correlation of the two formulae for the Hertzian stress and the shear stress will now be considered. From the former it is seen that P, the tooth load per mm, is proportional to the square of the Hertzian stress. The shear stress on the other hand is also proportional to the Hertzian stress, and as the admissible shear stress varies in direct proportion to the tensile strength, it follows that the admissible tooth loads vary with the square of the surface hardness.

A case-hardened, ground gear with a surface hardness of 56 Rc or 578 Brinell can, compared with an identical gear made of steel having 70 kg. per sq. mm. tensile strength and a hardness of 220 Brinell, be loaded $\left(\frac{578}{220}\right)^2 = 6.9$ times more, a fact proved by practice as will be indicated later. It must further be considered that hardening steels containing chromium and nickel show a higher resistance to dynamic loads and the above factor will be still higher when referred to static loading.

The gear calculation is further checked for scoring by the PVT formula. PVT is the product of the

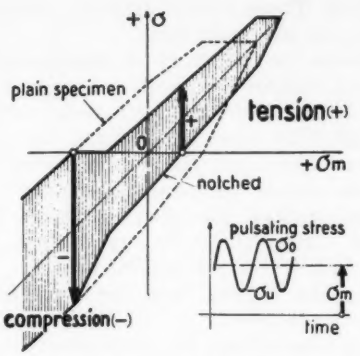


Figure 2. Goodman diagram.

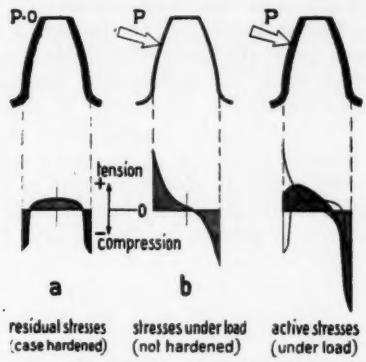


Figure 3. Effect of case-hardening.

Hertzian pressure in p.s.i., the sliding velocity in ft. per min. and the distance T along the line of action from the point of contact to the pitch point in inches.

BENDING STRESSES

As mentioned previously, the bending stress need not be considered for lightly-loaded conventional gears of carbon or alloy steel of 50 to 90 kg. per sq. mm. as the admissible load is limited to low values in order to avoid scuffing, scoring and pitting. The situation changes however for highly-loaded gears of alloy steel, such as hardened gears, for which the calculation of the bending stress may be as important as the one for the Hertzian pressure. It is therefore of interest to investigate the influence of hardening on the beam strength of the tooth flanks, more especially that of a case-hardened gear, where the hard uniform layer extends over the whole tooth flank and the fillet.

The Goodman fatigue load diagram shows the variation of the load from T_o to T_u at the right-hand bottom corner. The main diagram for the fatigue strength of a notched machine element (a gear tooth can be considered such a part) reveals that pulsating loads can be much better supported in the compression range than in the tension range. This means:

(a) that the compression side of a tooth can support, numerically, much higher stresses than the tension side, and

(b) that a considerable increase of the fatigue load could be realized if it were possible to displace the stress pattern of the loaded tooth from the tension range to the compression range by applying a system of favorable residual compressive stresses.

By case-hardening gears such a pattern of residual stresses is successfully obtained (Figure 3). By enriching the carbon in the carburized layer, compression stresses are raised, which are balanced by much lower tension stresses in the core. This residual stress pattern is characterized by high compression stresses. When loading a soft gear tooth a tension stress will be set up in the most highly-loaded section of the tooth on the side where the load is applied and a compressive stress will occur on the opposite side. In the case of a hardened gear under load, the dangerous tension stresses are considerably reduced by the residual compressive stresses, and the compressive stresses are higher on the compression side. In this way the desirable displacement of the tooth stresses from the tension to the compressive range, which, as the Goodman diagram shows, can be much higher loaded, is realized. The most favorable adaptation of the prevailing stress conditions to the effective load-carrying properties of a tooth is thus achieved.

COMPRESSIVE STRESSES

Residual compressive stresses can, of course, also be obtained by other methods such as nitriding, induction-hardening or shot-peening. Nitriding and shot-peening, however, do not produce sufficiently deep layers and consequently no deep-seated compressive stresses, while induction-hardening or flame-hardening often gives irregular distribution of the hard layer and therefore of the residual stress pattern.

ADVANTAGE OF CASE-HARDENING

Figure 4 draws a comparison between the active

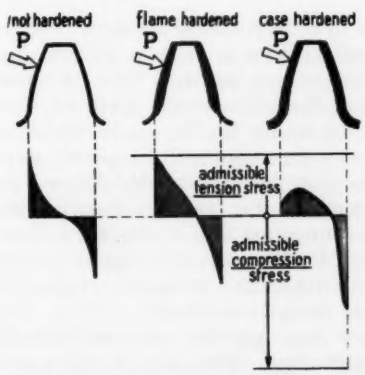


Figure 4. Active stresses under load.

stresses under load for three different types of gears. The first figure refers to the stress distribution of a soft gear, the second to a flame or induction-hardened gear (tooth-by-tooth method). The figure clearly shows the advantage of case-hardening, where in the highly loaded fillet there is a layer of uniform thickness as on the tooth flanks. The favorable residual stress pattern is therefore also present in the most highly loaded section of the tooth. The stress pattern of the single hardened teeth in the second diagram is not so favorable and, as shown in Figure 5, it may often be that the point of highest stress coincides with the point of lowest fatigue strength.

Extensive tests carried out for the Maag Gear-Wheel Co. by the Materials Testing Section of the Swiss Federal Institute of Technology at Zurich (EMPA) have clearly proved the statements just made on the favorable influence of case-hardening on the fatigue load.

One of the test devices, by means of which two opposed teeth are subject to a simultaneous and pulsating load, showed a pulsating fatigue strength of $T_B = 60$ to 65 kg. per sq. mm. for a case-hardened and ground gear of module 5. For a gear with very good flame hardened flanks, extending partly into the fillet, the corresponding value was 46 kg. per sq. mm. compared with 33 to 35 kg. per sq. mm. for a gear made of low alloy steel having a tensile strength of 75 to 80 kg. per sq. mm. Further interesting results of these tests are as follows:

Compared with a standard fillet a fully rounded-off tooth fillet cut with a protuberance cutter shows

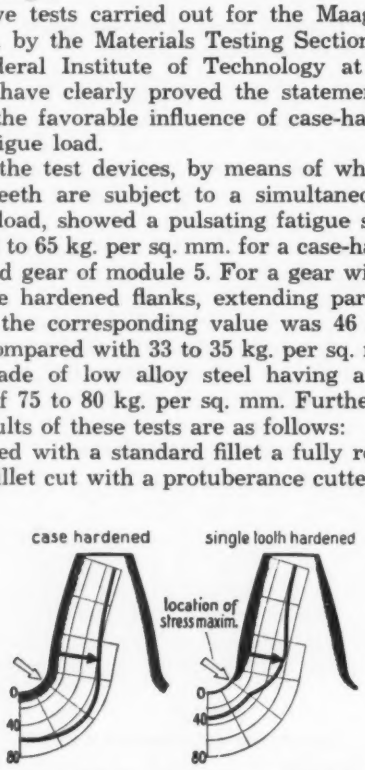


Figure 5. Distribution of effective fatigue strength.

an increase in strength of 47 to 58 per cent. A hardened untouched fillet is at least 10 per cent better than a hardened and polished fillet. It must be emphasized that the fillets of the teeth must never be ground, as otherwise the layers in which compressive stresses are present will be partly removed. It was always the practice of MAAG to leave the hardened roots of the teeth in their original state, and in this connection it is of interest to note that in recent years leading aircraft engine maufacturers have come to the same conclusion, whereas formerly they insisted on ground fillets.

The gears are usually pack-carburized, partly also gas-carburized. The case depth amounts for instance to 1 mm. for module 5 and 2 mm. for module 10-12. These values include a stock allowance of 0.3 mm. per tooth flank.

To conclude the summary of theoretical considerations and tests of hardened and ground gears it should be emphasized that the case-hardened layer of tooth and fillet must be uniform, and its thickness in accordance with the maximum depth of the shear stress aforementioned. The flanks must be ground with highest precision in regard to pitch, profile and concentricity on appropriate grinding machines. This execution allows gears of smallest dimensions, low weight, long life and highest reliability to be designed.

The Maag Gear-Wheel Co. commenced the manufacture of corrected involute tooth gears in 1910, and at the same time the manufacture of gear-cutting machines was taken up, to be followed shortly afterwards by the development of Maag grinders. Ground gears were gradually introduced for stationary gears, traction gears and other applications, forming the base of the firm's reputation.

All Maag gears are of the single helical type with helix angles between 6 and 15 degrees. The unavoidable small difference between the helix angle of the pinion and that of the gear-wheel is eliminated during erection by small displacements of the bearings. It is also possible to compensate for the torsional twist and bending of the pinion by an intentional misalignment.

Such correction cannot be made with a double helical design which is susceptible to axial vibration due to pitch and helix errors. In fact, it is never possible to match the two helix angles of the pinion exactly to those of the gear-wheel. Failures in double-helical marine gears, particularly in naval vessels were often caused by high-frequency vibration of the high-speed pinions. With double-helical gears any displacement of the main gear wheel due to an increase in the clearance at the thrust block can lead to premature wear or even tooth defects, whereas single-helical gears are insensitive to any such displacements.

By 1918, some gears transmitting 600 to 1,200 h.p. were supplied to France and Denmark, but it took many years to design and develop larger grinders for spur gears and later on also for helical gears. The first grinder for gear diameters up to 11 ft. 6 in., was put into service in 1932, and about a year later the first large marine gear was delivered to the Netherland Steamship Co. for the 10,500-ton cargo liner *Mapia*. Two 3,250 b.h.p. Sulzer engines running at 225 r.p.m., were coupled to drive a common propeller shaft turning at 86 r.p.m. The ship is still in service and the gear is in excellent condition.*

The French Navy was among the first customers who accepted peripheral speeds of 12,000 to 15,000 ft. per min. and recognized the advantages of hardened and ground gears. A considerable number of

* This was correct when the paper was read but she has very recently been scrapped.

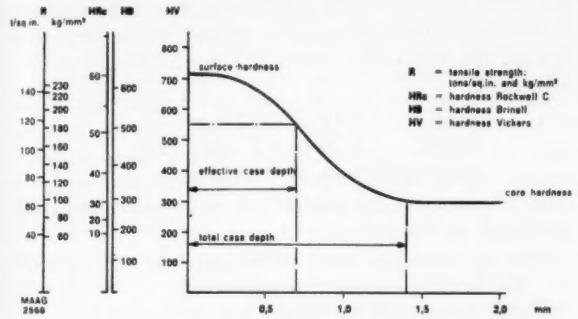


Figure 6. Hardness of tooth flanks in relation to case depth and core hardness.

gears were provided from 1935-1938 for auxiliaries on board the *Strasbourg, Richelieu, Mogador, Clemenceau* and others. These were for 1,800 to 3,380 h.p. with primary speeds of 7,500 to 8,500 r.p.m., at secondary speeds 1,200 to 1,700 r.p.m. The French destroyer *Agile*, for which Maag delivered the 14,000 s.h.p. main propulsion gears in 1939 was sunk before completion during the German occupation in 1940.

Before the Second World War, two marine gears of 7,500 s.h.p. each were supplied to the Belgian Navy for a coast-guard vessel. The Germans found the gears at Cockerills shipyard and had the propelling models thoroughly tested at Werkspoor Amsterdam, as they doubted that such a small gear would transmit this load. After extensive tests at full and overload the tooth flanks were still as new. This was one of the first gears for which air-hardening steel rims with a tensile strength of 140 kg/mm² were used instead of case-hardened rims. The main gear had a diameter of 5 ft. 6 in.

H.M.S. *Diana*

Shortly after the last war the U.S. Navy ordered two double reduction test gears of 6,600 s.h.p. each, which were to be tested by the back-to-back method. They carried a load of 256 per cent. The Admiralty also took great interest in hardened gears and the Daring-class ship *Diana* was equipped in 1951 with two Maag gear sets transmitting 27,000 h.p. each. These gears were back-to-back tested at Pametrada before installation on board.† These gears are of the dual-tandem articulated or locked-train type. The high speed pinions of the H.P. and L.P. turbines are each meshing with two primary gear-wheels each of which is coupled to the corresponding secondary reduction pinion by means of a quill shaft. The power is therefore transmitted to the main gear-wheel by four secondary reduction pinions. *Diana* has been in service since 1953 and the gear teeth are in perfect condition and do not show any pitting, scoring or scuffing.

† Pametrada Progress report 1951, "Marine Engineer and Naval Architect," 1952.

Y100 AND LATER SETS

The Royal Canadian Navy decided to equip their new destroyer escort vessels, fitted with Y100 turbines, with Maag gears.‡ All pinions as well as the first and second reduction gear-wheels are case-hardened and ground. Extensive trials were carried out with the first two sets of gears at Pametrada in 1952. Maag supplied six gears, each transmitting 15,000 s.h.p. for three destroyers, the first of which, the *St. Laurent*,‡ was commissioned in 1955. The gears for the further ships have been manufactured in Canada where a gear shop is equipped with Maag gear cutting and gear grinding machines, four of which have a capacity of 11 ft. 10 in. dia.

A 12,000 h.p. hardened and ground gear supplied by Maag was back-to-back tested by Avgra up to 18,000 h.p. on one flank and up to 26,000 h.p. on the other, with excellent results. The K-factor was as high as 1,253 on the test gear and 742 on the loading gear, striking proof of the theoretical statement made earlier, that a case-hardened and ground gear can be loaded at least 6.9 times more than a conventional gear. A set of 3 D.P. gears has been run at loads of up to 1,350 K-factor without damage of any sort.§ Further tests showed that tooth loading can be approximately doubled by suitable helix correction, compensating the twist.

The 30,000 s.h.p. single-shaft YEAD-1 experimental set, developed by Yarrow & Co. Ltd. for the Admiralty is based on the excellent results with the aforementioned Maag gears for the destroyers *Diana* and Y-100, and on the AVGRA tests. It was supplied by Vickers-Armstrong (Engineers) Ltd. and is of the locked-train type. All pinions and gears for both first and second reduction are case-hardened and ground on Maag machines. The K-factors are over 450 for both first and second reduction, compared with about 275 for a hobbed and shaved experimental gear and about 100 previously used in British naval gears.¶

The first of the new B-class Dutch destroyers, the *Friesland*, of 64,000 s.h.p. has two sets of Maag gears and the rest of the class have gears cut and ground on Maag machines by a Dutch shipyard.

GAS TURBINES

Modern gas turbines run at high speeds and have small dimensions, the corresponding gear must therefore run quietly at high peripheral speeds and also be of compact design. These requirements are easily met by hardened and ground gears. The French Navy use a large number of Maag gears in their free piston gasifier coastal minesweepers. The normal output is 900 s.h.p. at 500 r.p.m. per shaft, with an input speed of 9,003 r.p.m. The K-factor of the first reduction amounts to 203, and for the second reduction to 230. An escort vessel equipped with two gas turbines of 8,000 h.p. each and with Maag gears is under construction.

§ Transactions of the Institute of Marine Engineers, March 1957.
‡¶ Marine Engineers and Naval Architect Annual Steam Number 1956, London.

MERCHANT SHIP APPLICATION

To begin with, the introduction of hardened and ground gears for merchant ships, in fact the largest field of application, met with considerable difficulties. The rules of the different Classification Societies existing after the Second World War limited the K-factors for gear calculations to values which were safe for soft hobbed gears. However, splendid results achieved with heavily-loaded Maag gears in stationary applications, led Lloyd's Register, in 1948, to agree to higher K-factors. The teeth of these gears after 20-25 years' service were practically in the same state as when delivered.

For double-reduction main propelling machinery, with quill shafts, a K-factor of 284 is permitted if both pinion and gear are case-hardened to 56 Rockwell and ground, as shown in Figure 7. If a case-hardened pinion is used with a gear made of oil-hardened or air-hardened nickel-chrome steel with a tensile strength of 125 to 140 kg/mm², the permissible K-factor is 213. The third case refers to a case-hardened pinion and a gear made of heat-treated nickel steel having a tensile strength of 80 to 95 kg/mm², with an allowable K-factor of 142. For conventional hobbed gears the factor amounts to 80 only. The next example gives the corresponding values for designs without quill shafts. Gears for auxiliaries may be loaded to K-factors of 318, 238 and 159 respectively, whereas the figure for soft hobbed gears is 90. The other Classification Societies soon fell into line and accepted similar values.

For case-hardened pinions, nickel chromium steel having the following chemical composition is generally used:

carbon	0.12 to 0.17 per cent.
nickel	3.2 to 3.7 per cent.
chromium	0.55 to 0.85 per cent.

The mechanical properties of a longitudinal test piece are as follows:

Tensile strength min. 80 kg/mm² in the core of the teeth

Yield point = 60 kg/mm²

Notched bar impact strength on 10 mm² test bar longitudinal: min. 8 kg/cm²

Surface hardness of the tooth flanks min. 56 Rockwell C.

For naval vessels the first reduction rims are generally made of the same material. In the case of merchant vessels, the first and second reduction gear-wheels consist of a body made of cast steel or cast iron with shrunk-on rims of oil-hardened Ni-Cr-Mo steel. These rims are delivered already heat-treated by the steel-maker before any machining is done. Such steels can easily be cut on Maag gear-generating machines by means of rack-shaped cutters, and in spite of the high hardness clean tooth flanks are obtained. Grinding is, however, necessary for the high-accuracy finish. According to the tooth loading the rims are heat-treated to 90 to 105 or 125 to 135 kg/mm². The former steel contains approximately 0.3 per cent carbon, 3 per cent nickel and 0.8 per cent chromium, whereas the composition of the latter is approx. 0.3 per cent carbon, 4 to 5 per cent nickel and 1.2 per cent chromium.

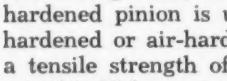
CORRECTIONS APPLIED WHEN GRINDING

The teeth of marine gears are always cut with protuberance cutters. Undercutting of the tooth flanks is thus obtained, and therefore the ground tooth profile blends nicely into the fillet. All gear teeth are finished by grinding, with tooth-to-tooth pitch errors not exceeding a few thousandths of a millimeter. The latest types of Maag gear grinders enable any desired tooth corrections to be carried out on highly-loaded gears. The teeth are ground with tip and root relief, and also with partial longitudinal crowning. With Maag machines it is possible to obtain a tip and root relief of up to 0.05 to 0.08 mm. and a longitudinal crowning up to 0.03 mm., fully automatically. The same corrections can be reproduced any time later on. The crowning device also enables helix angle corrections to be ground.

A REPLACEMENT SET

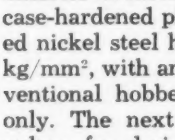
Shipowners are rather conservative, and it took some time to interest some of them in gears with hardened and ground teeth. Eventually Maag received an interesting order for the replacement of the rotating parts of an articulated double-reduction gear with double-helical teeth, for the Victory-ship

I. REDUCTION GEARING FOR MAIN PROPELLING MACHINERY FOR SINGLE AND DOUBLE-REDUCTION GEARS



a With quill shaft

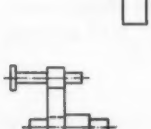
	Lloyd's Register factors for: Maag gears			Conventional gears
	I	II	III	
1st and 2nd reduction:	284	213	142	80



b) Without quill shaft

	Lloyd's Register factors for: Maag gears			Conventional gears
	I	II	III	
1st reduction:	284	213	142	80
2nd reduction:	241	181	120	67

II. REDUCTION GEARING FOR AUXILIARY ELECTRICAL GENERATING MACHINERY



Lloyd's Register factors for: Maag gears			Conventional gears
I	II	III	
318	238	159	90

Figure 7. Tables showing permissible K-factors when hardened and ground gears are used in various applications.

Molenkerk of the United Netherlands Navigation Co. The replacement gears are of single-helical type, all pinions, the wheels of first reduction having case-hardened and ground teeth, the second reduction wheel having a shrunk-on rim of heat-treated 60-ton alloy steel with ground teeth. As the existing gear case remained in Amsterdam, the new pinions and gear-wheels could not be tested before despatch from Zurich. The normal power transmitted is 6,600 h.p., with primary speeds of 5434 and 4021 r.p.m., and propeller shaft speed of 100 r.p.m. The K-factors of the Maag gears are 226 and 177 for the first reduction and 135 for the second reduction. The corresponding values for the original design were 94 and 72 for the first reduction and 70.5 for the second reduction. There was a striking reduction in the face width of the gear teeth, from 2×190 mm. to 160 mm. in the first reduction and from 2×446 mm. to 460 mm. in the second reduction. The ship has been in regular service since 1953, and a year ago the owners stated that after 200,000 sea-miles the gears showed no wear or pitting and that the magnetic filters remained free from metal particles.

The first two complete gear sets with single-helical teeth for merchant ships were installed in 1954 in the 25,000-ton tankers *Tarim* and *Proteus* to suit three-casing Parsons-type turbines built under license in the Netherlands. Each gear set transmits 10,890 s.h.p. at 111 r.p.m. The pinions as well as the three rims of the first reduction are case-hardened and ground. There is no flexible element such as a quill shaft between the primary and secondary trains. The rims of the first reduction are bolted to the discs and the latter are fixed on the second reduction pinions in the same manner. The casing is made of cast iron. The weight of the entire gear set is 55 tons which compares with at least 100 tons for a contemporary conventional hobbed gear.

MORE GEARS FOR TANKERS

In 1956, three 18,700-ton tankers of the Atlantic Refining Co., Philadelphia were fitted with hardened and ground gears, transmitting a total of 10,500 h.p. from the 5,445 and 5,330 r.p.m. turbine shafts to the propeller shaft at 103 r.p.m. Maag supplied the rotating parts, and the ship and engine-builders, Cockerill in Belgium, built the casings. The first-reduction gear-wheels consist of cast-steel bodies with shrunk-on rims, a construction used in all subsequent Maag gears for merchant vessels. Quill shafts connect the primary gear-wheels with the second reduction pinions. The bull gear is of cast iron with a shrunk-on nickel chrome rim heat-treated to 110 kg/mm^2.

For powers of about 15,000 h.p. and above, the smallest dimensions and lowest weights are obtained by using the locked train configurations. A Maag gear recently delivered for a 38,000-ton tanker, transmits 19,250 h.p. at 101 r.p.m. with a total weight of only 83 tons.

Many reduction gears for use between steam turbines and auxiliary generators on board ship have also been supplied with case-hardened pinions and gear-wheels, these covering powers ranging from 400 to 1,500 kW. Hardening and grinding lead to a compact design which saves space. The turbines run at 9,000 to 10,000 and the generators at 1,200 to 1,800 r.p.m.

GEARED DIESEL APPLICATIONS

Hardened and ground gears are of particular interest for geared diesel installations. Backlash between the teeth can be minimized due to the high accuracy of grinding, and hammering between the teeth is thus avoided. Further, the hard surface of the teeth is able to withstand considerable overload due to torque variations and torsional vibrations. Solid coupling between engine and gear is therefore in many cases possible, the torsional stress calculations being much more reliable than those for systems with a flexible element.

The crankshafts of two diesel engines each developing 850 h.p. at 275 r.p.m. are rigidly coupled to the pinions of a twin-gear set delivered in 1942. The propeller shaft turns at 110 r.p.m. In 1949 ten cargo vessels were equipped with Maag gears rigidly coupled to 1,700 b.h.p. at 275 r.p.m. Werkspoor diesel engines.* Today the tooth surfaces are still as though the gears had not been run at all.

SUMMARY

The results so far obtained with these gears can be summarized by stating that hardening and grinding lead to minimum dimensions, low weight and highest safety. Such gears do not show any signs of pitting, scoring or scuffing, even after long years of service.

In this connection an investigation carried out by Lloyd's Register on gear failures and covering the period 1953-1956 is of interest. All these gears are of the double-helical type with soft hobbed teeth. Out of all classified ships above 3,000 tons d.w., 83 had gear failures. Scuffing and pitting were responsible for 55 per cent of the defects and only 20 per cent were due to tooth fractures.

During the past 18 months, gear failures of a particular nature have occurred in several European-built ships, particularly tankers. The damage generally consisted of premature wear on the tooth flanks of the bull gear from the pitch line to the root of the teeth. These gears were also all of the double-helical type with soft hobbed teeth. Severe erosion on the tooth flanks was experienced, sometimes reaching over 0.5 mm. within a few months and even tooth breakages occurred. The defects were partly attributed to bad material, and vibrations in the stern of these ships.

* "Marine Engineer and Naval Architect," April and December 1951.

Recent investigations however revealed that longitudinal compression vibrations of the propeller shaft, so that the bull gear transmits the load alternatively by the left and right hand helices, with consequent heavy overloading of the tooth flanks, must be the cause. These longitudinal compression vibrations also give rise to forced torsional vibrations in the gear. With single-helical gears the load on the tooth flanks is not influenced by the compression vibrations, and the forced torsional vibrations are much lower, as due to the smaller helix angle the tangential amplitude on the bull gear is much less.

Mr. G. J. O'Donnell of the Shell Development Company has submitted a possible explanation for an observed phenomenon in spur gear operation at high speeds. The anomaly is that, as the spur gear speed is increased beyond a critical point, the load carrying capacity ceases to decrease and begins to increase. Tooth pressures as high as 500,000 psi have been observed, yet it is indicated that lubricants have been able to prevent metal-to-metal contact under certain conditions.

It is hypothesized that the lubricant possesses a property of viscoelasticity which enables it to accomplish this result. Viscoelasticity is the property of a fluid to show an elastic response to a very high rate of application of sheer stress. It has been shown that all liquids, (and even air) possess this property. Mr. O'Donnell explained the relationships involved and derived a time constant called the "relaxation time" which is a characteristic of each particular fluid depending on the viscosity and shear modulus. He then postulates that when the time of stress application is in the order of the time constant, the lubricant behaves elastically and is able to resist extremely high loadings. Further investigation into this property of lubricants is urgently needed.

—from MECHANICAL ENGINEERING, February, 1959

The rocket age has introduced a new environmental hazard to mechanical and electrical equipment reliability. Noise in the range of 160 to 170 decibels can cripple components which can readily pass ordinary mechanical vibration tests. In order to test such components, the Avco Research and Advanced Development Division has developed an electromechanical loudspeaker which can produce up to 163 decibels of random noise, or up to 170 decibels at a selected frequency. The noise generator, incorporating a 12 inch diameter moving coil, is used with a progressive wave tube or a reverberant chamber. In the former, the noise is beamed down the tube, which contains the component, and is absorbed by damping material at the opposite end. Orientation of the component within the tube permits a selected directional impingement of the sound energy on the test article. The reverberant chamber is designed for larger components and reflects the noise from all surfaces. Input power to the generator is 3 KW.

—from THE JOURNAL OF THE FRANKLIN INSTITUTE, February, 1959

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THE INFLUENCE OF LAW AND INTERNATIONAL TREATIES ON THE DUTIES AND SHIPS OPERATED BY THE COAST GUARD

THE AUTHOR

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In 1949, Congress reviewed the duties and functions which had accrued to the Coast Guard since its inception in 1790. These duties were amended, revised, codified, and enacted into one law, Title 14, United States Code. For the first time since 1790, there existed clear, concise, and consolidated statutory authority for all the Coast Guard's many functions.[1] These may best be summarized by Section 2 of Title 14, United States Code, entitled "Primary Duties."

"The Coast Guard shall enforce or assist in the enforcement of all applicable Federal Laws upon the high seas and waters subject to the jurisdiction of the United States; shall administer laws and promulgate and enforce regulations for the promotion of safety of life and property on the high seas and on waters subject to the jurisdiction of the United States covering all matters not specifically delegated by law to some other executive department; shall develop, establish, maintain, and operate, with due regard to the requirements of national defense, aids to maritime navigation, icebreaking facilities, and rescue facilities for the promotion of safety on and over the high seas and waters subject to the jurisdiction of the United States; and shall maintain a state of readiness to function as a specialized service of the Navy in time of war."

The duties of the Coast Guard may be classified in four categories. These are Law Enforcement, Safety at Sea, Aids to Navigation, and Military Readiness. Before exploring each of these, a brief history of the Coast Guard will be included to demonstrate the influence of law on the beginnings of this service.

EARLY HISTORY

On April 23, 1790, Alexander Hamilton, Secretary of the Treasury, requested an appropriation from Congress to build ten revenue cutters with which to enforce the Revenue Act of 1789. On August 4, 1790, Congress concurred, and established the Revenue Cutter Service with its primary duty of law enforcement.[2]

When the Federal Government was established under our present Constitution in 1789, among the duties it assumed from the separate states was that of operating a system of Aids to Navigation. At that time, the system consisted of twelve coastal lights and seven buoys. This system, sometimes called the Lighthouse Service, was placed under the Secretary

of the Treasury, and was the forerunner of the present Aids to Navigation Section of the Coast Guard.[3]

In 1831, John McLane, Secretary of the Treasury, detailed seven of the revenue cutters to patrol areas near their stations during the winter months for the purposes of rendering assistance to vessels in distress, and of saving life and property. In 1837 Congress authorized the President to employ public vessels for the purposes of cruising the coast during severe weather, and of affording aid to distressed mariners. This was the beginning of the Coast Guard's Search and Rescue Section, an important part of Safety at Sea.[4]

After Robert Fulton's *Clermont* was successfully operated in 1807, steam vessels became more common on United States waters. Along with steam came fires and disastrous explosions. In 1832 alone, 14 per cent of the steam vessels then operating were destroyed by explosions and fires, and over one thousand persons lost their lives. To cope with these disasters, Congress, in 1838, created the Steamboat Inspection Service and with this law the Merchant Marine Inspection and the Merchant Marine Safety phases in our present Coast Guard were born.[5]

From the foregoing history it should be noted that the statutory skeleton of the Coast Guard was established at an early date. Consolidation, additional duties, and changes necessitated by technological progress have been built upon this framework.

LAW ENFORCEMENT

The Coast Guard today is the primary Federal Law Enforcement Agency afloat, having law enforcement responsibility on all waters over which the United States exercises jurisdiction and for all United States vessels on the high seas. Section 89, Title 14, United States Code is the statutory authority. The Coast Guard is primarily responsible for enforcing the following laws.

1. *Prevention of the illegal landing of dutiable goods, of the smuggling of articles completely prohibited in the United States.* This is the law for which the Revenue Cutter Service was established in 1790. In addition to the regular oath of members of the Armed Services, it is interesting to note that all Coast Guard officers take an oath to protect the revenue of the United States. In this connection, Coast Guard officers may board any vessel over which they exercise jurisdiction to examine its manifest and cargo for contraband, and to check its documents for fraudulent operation.
2. *Enforcement of navigation laws*, among which are the International Rules, Inland Rules, Great Lakes Rules, and Western Rivers Rules. The International Rules and revisions thereof are examples of an international agreement ratified by our Congress. The Coast Guard sends representatives to any conference considering the revision or amendment of these laws, and Coast Guard officers were among United States delegates to the last International Conference on Safety of Life at Sea held in London in 1948. Other minor navigation laws enforced by the Coast Guard include the regulations for anchorages established by the Secretary of the Army, the Oil Pollution Act, laws pertaining to the obstruction of the Aids to Navigation, and the obstruction of navigable waters. An increasing number of off-shore structures, especially in the Gulf of Mexico area, has required considerable Coast Guard surveillance to insure proper markings of the structures, and that safe operating procedures are being employed.
3. *Enforcement of Marine Inspection laws.* This subject will be dealt with in detail under the category of Safety at Sea.
4. *Enforcement of motor boat laws.* With the growing number of pleasure craft on the waters this has become an important phase of Coast Guard operations. Many people are operating a boat for the first time with little knowledge of seamanship, weather, safety, navigation, and the like. These people are a menace to themselves and to the qualified operators. It seems reasonable to suppose that within a few years, all operators will be required to possess some sort of maritime license. At present only operators that carry passengers for hire are required to be licensed.
5. *Patrolling of marine regattas, parades, races, or spectacles.* Here the appropriate district commander establishes the regulations for each event and district units enforce these regulations.
6. *The transportation of the federal "Floating Court" annually from port to port in Alaska.* This brings the courthouse and justice to coastal communities in Alaska that are too small or too inaccessible to have a permanent court. Doctors and dentists are also transported to attend to anyone requiring their services. Wherever necessary, Coast Guard officers may be appointed as United States Commissioners and United States Marshals in Alaska.
7. *Protection of marine life.* In most cases this duty is as a result of a treaty with a foreign nation. The treaties for which the Coast Guard is the primary enforcing agency, or cooperates with the Fish and Wildlife Service of the Department of the Interior are:
   a. Fur Seal and Sea Otter in Northern Pacific waters
   b. Northern Pacific Halibut Fisheries
   c. Sponge Fisheries in the Gulf of Mexico
   d. Whaling Convention of 1949
   e. Tuna Convention of 1950
   f. Northwest Atlantic Fisheries[6]
8. *The rendering of assistance to any federal agency, state, territory, possession, or subdivision thereof.* This assistance is usually of a law enforce-

ment nature such as the assistance given the Fish and Wildlife Service in enforcing many conservation laws. Another example is enforcement of the immigration laws in cooperation with the Immigration Service. Assistance other than law enforcement include the operation of a Voice of America mobile relay station, in this case a ship, at Rhodes, Greece, for the State Department. The State Department also requested the establishment of the Campeche Patrol to protect United States shrimpers in waters near Yucatan. Close cooperation is maintained with the Department of Defense with respect to Aids to Navigation matters.

9. *The enforcement of dangerous cargo and port security laws.* Under the authority of the Magnuson Act of 1950, the President by executive order designated the Coast Guard as the federal agency responsible for prescribing and enforcing the regulations for the handling of dangerous cargo, safety measures on vessels at waterfront facilities, vessel movement, and the security of the waterfront in specified areas. For these duties the Coast Guard established port security units. Under the section on Military Readiness, these will be discussed in more detail.[7, 8]

As a result of the law enforcement work legislated on the Coast Guard, its officers must be familiar with such phases of the laws as jurisdiction, arrest, offense, warrants, boarding, search, seizure, territorial waters, customs waters, treaty vessels, high seas, hot pursuit, and so forth. Its officers must also know the applicable laws as found in the United States Code, the United States Code Annotated, the Federal Register, the Bill of Rights, and the Constitution. Failure to be familiar with these may result in an illegal arrest or seizure of a citizen for which the United States may be sued, and for which the offending officer may be disciplined.

SAFETY AT SEA

Safety at Sea, the next phase of Coast Guard duty to be discussed may be subdivided into Search and Rescue, International Patrols Required by Treaties, and Merchant Marine Safety and Inspection.

The winter cruising established in 1837 was an annual feature of Coast Guard work until 1941 when it was interrupted by war. After the war, the establishment of Search and Rescue facilities at key coastal points, together with progress made in communications, made the cruising system obsolete.[9]

In the period from 1837 to 1941, several important things occurred with regard to the Coast Guard's work on Safety at Sea. In 1871 the Life-Saving Service was established as a separate Treasury Department Bureau for the purpose of saving life and property along the coast. This was a service separate from the Revenue Cutter Service, but Revenue Cutter Service officers acted as inspectors and drill masters. In 1915, these two services were joined and the name changed to the United States Coast Guard.[10] The functions which were performed by the Lifesaving Service, namely surf work and the operation of lifeboat stations, seemed to characterize the Coast Guard in the eyes of the average layman.

Since World War II, great strides have been made in coordinated air-sea search and rescue teams. Cooperation with other countries became important with the expansion of commercial air traffic. Coast Guard's aviation arm pioneered in the use of the helicopter as an important aid in rescue. The Coast Guard also gave the benefit of its experience in off-shore landings to the commercial airlines. Ditching procedures have prevented the loss of many lives.

The importance of International Treaties on Safety at Sea became especially evident with the establishment of the International Ice Patrol. The United States was designated the operating government, and our government designated the Coast Guard to be its operating agency. The Coast Guard has made increasing use of aircraft in the patrol. The surface vessel's role has become more oceanographic in nature, although on occasion surface vessels remain in the area of a berg if it is a hazard to navigation.

Another important international group is the International Civil Aviation Organization (ICAO), established in 1945 with headquarters in Montreal.[11] This agency, to which the United States belongs, has established standards and areas of responsibility for search and rescue work throughout the world. In 1946, President Truman set up the Air Coordinating Committee to oversee United States policy with regard to ICAO.[12] This committee designated the Coast Guard as the responsible agency for establishing, maintaining, and operating search and rescue facilities on and over the high seas, waters subject to the United States jurisdiction, and adjacent land areas. The Continental Air Command of the Air Force is responsible for the inland areas of the United States.[13] The Coast Guard was the United States operating agency charged with complying with ICAO requests upon this country. Major amongst these requests was the establishment of floating ocean stations for the purpose of providing search and rescue facilities, communications, air navigation facilities, and meteorological services in designated ocean areas.[14]

ICAO also requested the establishment of rescue coordination centers by its member nations, and set forth certain minimum requirements and procedures in their operation.[15]

The commercial airlines, both United States and foreign flag lines, have benefited from ICAO, and all shipping has been aided by the more complete weather services made available as a result of reports sent in by ICAO's ocean weather stations. As commercial overseas aviation increases, these duties of the Coast Guard seem likely to grow rather than to contract.

MERCHANT MARINE SAFETY AND INSPECTION

The Steamboat Inspection Service underwent several reorganizations between its establishment in 1838 and World War II. In 1884, the Bureau of Navigation was established to deal with the problems of maritime personnel, and to collect fees for the issuance of licenses and certificates of inspection of vessels. Positions as Shipping Commissioners were established to perform these tasks.

In 1903, the Bureau of Navigation and the Steamboat Inspection Service were transferred from the Treasury Department to the Department of Commerce where they remained for thirty-nine years. In 1936 these units were united to form the Bureau of Marine Inspection and Navigation. This bureau joined the Coast Guard for the duration of World War II in 1942 by presidential executive order. The President made this transfer permanent in 1946 and included the following functions of Merchant Marine Safety and Inspection under the Coast Guard.

1. Insure that vessels are of a structure suitable for the service in which they are to be employed.
2. Insure that vessels have suitable accommodations for passengers and crews.
3. Insure that vessels and their equipment are in suitable condition to warrant the belief that they may be operated safely.
4. Insure that vessels are suitably equipped and adequately manned to ensure their safe navigation and operation, and to provide for safety of life of their passengers and crews in the event of fire, collision, stranding, sinking, or other marine casualty or disaster.

To perform the foregoing the Coast Guard must:

1. Approve plans, materials, equipment, and appliances for the construction, alteration, and repair of vessels. This required the establishment of a Merchant Marine Technical Division in Coast Guard Headquarters which must be staffed with competent officers. M.I.T. course XIIIA graduates are presently filling these billets.
2. Inspect vessels, their equipment, and appliances. Specific statutes exist as to the frequency of inspection, and those vessels required to be inspected.
3. Issue certificates of inspection, and permits indicating the approval of vessels for operations which may be hazardous to lives or property. This refers to the Coast Guard's regulations for the handling and storage of dangerous cargo aboard vessels and at waterfront facilities.
4. Administration of loadline requirements. Thirty nations met and established the requirements in 1930 in London. This agreement was ratified by Congress in 1931.
5. License and certify officers, pilots, and seamen, and suspend them when necessary.
6. Investigate marine casualties.
7. Enforce manning requirements, citizenship requirements, and requirements for the mustering and drilling of crews.
8. Supervise the shipment, discharge, protection, and welfare of merchant seamen.[16]

INTERNATIONAL TREATIES

The influence of international treaties has been very important in the Safety at Sea phase of Coast Guard operations. In addition to the International Load Line Convention, there have been several International Conventions for Safety of Life at Sea. The first major one, in 1913, recommended certain minimum standards in subdivision of passenger ships, boats, and lifesaving appliances. A direct result of this early convention was the establishment of an International Ice Patrol.[17]

A second International Conference was held in 1929 and the most recent in 1948. The provisions of the 1948 convention included the setting up of uniform practices amongst the signatory nations in inspections, surveys, and casualty investigations. In new construction it provided requirements in subdivision and stability, fire protection, pumping arrangements, electrical installations, maneuvering power, and steering. It also included requirements in lifesaving appliances, emergency drills and procedures, radiotelegraphy and radiotelephony aboard ship, safety of navigation, carriage of dangerous cargo, and a revised set of the International Rules of the Road. Coast Guard officers were members of the United States delegation to this conference.[18]

Coast Guard officers assigned to Merchant Marine Safety and Inpection are required to keep abreast of new developments affecting this field, and to initiate new or changed regulations as they seem necessary. The forthcoming construction of an atomic powered merchant vessel is an example of an entirely new area which must be explored by the Coast Guard in conjunction with other interested agencies.

Established at the 1948 Conference was the Intergovernmental Maritime Consultative Organization, IMCO. Our Congress ratified IMCO in 1951 and designated the Coast Guard as one of several United States agencies to take part in its functioning.

AIDS TO NAVIGATION

As our foreign commerce grew, so did the Lighthouse Service's responsibilities. It remained a separate service until 1939 when President Roosevelt made aids to navigation a duty of the Coast Guard.

The Coast Guard then had the responsibility of the country's lighthouses, buoys, lightships, radiobeacon stations, and associated maintainance facilities. It has been extremely fortunate that in its acquisition of these separate agencies (Lighthouse Service, Bureau of Marine Inspection and Navigation, and Lifesaving Service), the Coast Guard has also integrated the personnel of these bureaus into

its own military organization. In most cases the integration amounted to the change of a uniform and the taking of an appropriate oath of office. These personnel have remained within their own specialties until their retirement. They have been invaluable in training the regular Coast Guard general duty officer in their particular specialty.

The most significant development in aids to navigation work has come as a result of technological advances in electronics. The Coast Guard now operates a chain of Loran stations that practically provide world wide coverage. Section 81, Title 14, United States Code gives the Coast Guard authority to establish, maintain, and operate aids to navigation to serve the commerce and the armed forces of the United States. Under this authority the Defense Department has requested the Coast Guard to establish Loran stations and other aids to navigation in areas one would hardly expect the Coast Guard to be operating.

Most of the Coast Guard's research and development work is carried on in the field of aids to navigation. For this work an electronic test station and a testing and development unit for non-electronic items are being operated. Work is conducted on improving Loran timers and transmitters, radar devices, marine paints, improved lens and acoustics systems for lighthouses, and many more.

A corollary duty to aids to navigation is icebreaking in harbors and channels to meet the reasonable demands of commerce.

MILITARY READINESS

The Coast Guard stands ready to become part of the Navy should a state of war be declared, or by presidential order. To facilitate such a change, the Coast Guard maintains its files, correspondence, organization, and the like, similar to that of the Navy. Larger Coast Guard floating units undergo Navy refresher training every two years. All Coast Guard directives are issued with the idea of possible operation under the Navy at any time.

Revenue Cutters first participated in military action in the Quasi-War with France in 1799. Since then, they have been in all major and minor wars fought by the United States. In World War II, in addition to operating their own vessels, Coast Guard officers and men operated destroyer escorts, frigates, corvettes, PC-boats, transports, attack transports, attack supply vessels, and many landing and amphibious craft.[19]

The second phase of military readiness is the Port Security Program. The passage of the Magnuson Act in 1950 allowed the President to institute port security measures by executive order without declaring a national emergency. Such an order was issued shortly thereafter, Executive Order 10173, with local authority vested in the Captains of the Port.

The responsibilities of the Port Security organization within the Coast Guard include the following:

1. The prevention of the illegal or hidden entry from sea of persons, vessels, or things which threaten the safety and security of the United States. At certain ports, harbor entrance patrols are maintained, and all vessels entering are thoroughly checked. Special attention is paid to Iron Curtain vessels including inspection with Geiger-counters.
2. Issuance and checking of credentials required of persons in certain waterfront areas. The office of Merchant Marine Safety at headquarters takes care of security checks and clearances for Merchant Marine personnel, along with other documents.
3. Prescribing and enforcing conditions and restrictions relating to the safety of waterfront facilities, vessels, and anchorages. This particularly includes the loading and handling of dangerous cargo.
4. Complete control of all vessel movements and anchorages whenever conditions warrant.
5. During wartime or national emergency, the placing of armed guards at key waterfront areas and coastal points.[20]

The preceding material has shown how law and treaties have affected the Coast Guard's authority and operations in the four main categories of law enforcement, safety at sea, aids to navigation, and military readiness. These duties are summarized in Appendix I. The trend over the years has been one of growth and consolidation directed especially at bringing all federal interests relating to the sea, and commerce thereon, under one department. With increasing emphasis being given to the United Nations and to regulation of the many avenues of international intercourse, the Coast Guard is the likely federal agency to be responsible for this country's participation and compliance with maritime affairs.

EFFECTS OF DUTIES ON COAST GUARD SHIP DESIGN

Ships required to perform these varied duties have evolved over the years. For its long distance work such as International Ice Patrol, ocean stations, Alaska patrols, long range search and rescue work, and various wildlife patrols, the Coast Guard requires a ship with good cruising radius, moderate speed, fairly comfortable riding qualities, good stability under adverse conditions, arrangements for towing and assistance work, provisions for a great amount of electronic gear, and some armament.

It may be noted that the present ship performing this duty, the cruising cutter, is about 300 feet long, 40 feet in beam, and has a speed of about 18 knots. The beam is larger than that found on a commercial ship of similar length, thus adding to the stability, an important factor in rescue work. Other characteristics of the cruising cutter include a flush deck, bow sections with pronounced flare, fairly straight

midships sections with medium freeboard for ease in small boat operations, goodly use of shear, a cruiser type stern, and a clear fantail for towing.[21] Examples of cruising cutters now employed are: Figure 1, a 327′ gunboat; Figure 2, a seaplane tender, small; Figure 3, a 255′ gunboat. Military readiness, with regard to operating as a part of the Navy, is important to these ships. They are the only Coast Guard ships undergoing Naval refresher training.

U.S. Coast Guard Official Photo
Figure 1.

For shorter range duty such as yacht races, coastal rescues, patrols, law enforcement and port security work, a slightly different type of ship is required. This type is usually a smaller version of the cutter, but maintains the same general shape and lines. Examples of this type of craft are: Figure 4, 165′ patrol craft; Figure 5, 125′ patrol craft; Figure 6, 95′ patrol boat; Figure 7, 83′ patrol boat. The major difference occurs in the propulsion plant design which is most affected by the shorter required cruising radius, the need for ease and speed in getting underway, and maneuverability.

Another class of cutters which can perform law enforcement and safety at sea work, but which is designed for a different purpose, is the icebreaker. The Coast Guard operates three of these along the coasts and one on the Great Lakes. Figure 8 shows a Wind class icebreaker. The icebreaker not only

U.S. Coast Guard Official Photo
Figure 2.

U.S. Coast Guard Official Photo
Figure 4.

U.S. Coast Guard Official Photo
Figure 3.

U.S. Coast Guard Official Photo
Figure 5.

U.S. Coast Guard Official Photo
Figure 6.

U.S. Coast Guard Official Photo
Figure 7.

U.S. Coast Guard Official Photo
Figure 8.

keeps harbors open to commerce, but has proven indispensable on the annual logistics trips to northern military outposts and on voyages to the Antarctic.

The usual armament found on the Coast Guard's cruising and patrol cutters is a surface firing gun of five inch maximum size, an anti-aircraft weapon such as the 40 mm, and anti-submarine weapons consisting of the depth charge and a rocket-type weapon. All armament is furnished the Coast Guard by the Navy.

Two distinct types of ships are required for aids to navigation work. These are the tender and the lightship. The tender also performs search and rescue, law enforcement, and local icebreaking duties. The tenders have broad beam, relatively shallow draft, large unobstructed well deck forward, powerful hoisting equipment, large boom, and large water and fuel tanks to perform logistics work. An example of a tender may be seen in Figure 9, a 180′ C-class tender. The broad beam enables the tender to carry heavy deckloads, and to lift and handle buoys over the side even in the open seas. The majority of tenders are of diesel electric propulsion with pilot house controls. The speed of the average tender is between 10 and 15 knots. They are about 180 feet long for outside coastal work.[22] Many tenders have a specially designed bow and extra plating for icebreaking work in harbors and rivers. Military readiness is not as important to these ships as it is to the cruising cutters.

The lightship is the other special type of ship required in aids to navigation. Hull forms have evolved by trial and error. The hull has to be capable of withstanding the unusual conditions encountered while anchored fully exposed to the sea. Steadiness and ease of motion are the chief requirements for the proper display of the light, as well as for the comfort of the crew. Lightships are designed full above the waterline, forward and aft, to increase the vessel's displacement rapidly when pitching occurs. (See fig. 10.) Pronounced tumble home, bilge keels, and ballast are also used to reduce motion in a seaway. Metacentric height is low, and is of the order of one foot. Scantlings throughout are much heavier than are ordinarily required in vessels of similar size. Extra heavy duty anchor gear is required, namely a 7500 pound mushroom

U.S. Coast Guard Official Photo
Figure 9.

U.S. Coast Guard Official Photo

Figure 10.

anchor leading out through a stem hawse pipe. Power plants vary in type depending upon the location of the station and the average weather expected. These conditions dictate the amount of power required to get to and from the station, and to provide services while on station.[23] Considering the discomfort, danger, and isolation of the men on lightships, the Coast Guard is working on the development of an unmanned, radio-controlled ship to maintain station.[24]

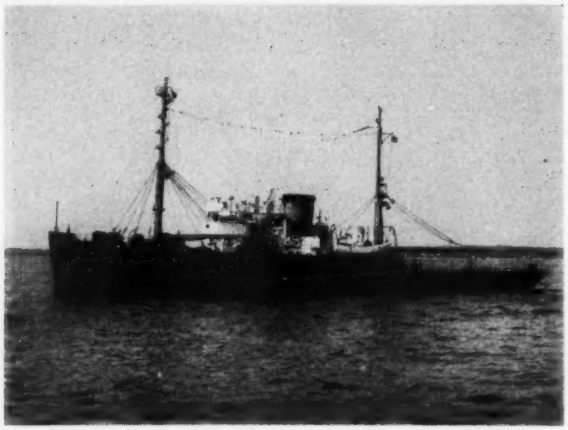
Several ocean and harbor tugs are also important members of the Coast Guard fleet. These make the ideal assistance vessel, and it seems probable that more will be acquired. See Figures 11 and 12.

Technical data on many Coast Guard vessels will be found in Appendix II.

Other special small craft are required to perform assistance, law enforcement, and port security work at lifeboat stations and small coastal units.

A large percentage of today's Coast Guard fleet is reaching replacement age. A sizeable construction program will soon be started and should carry on for a number of years. This construction period will be the first major one since the amalgamation of the Lighthouse Service and the Bureau of Marine Inspection and Navigation. New ships will probably be similiar in lines to their predecessors since the important characteristics affecting their design have not materially changed. Changes will occur in habitability, electronics, and ordnance. Propulsion plants will be modernized and probably will be largely diesel, but may possibly include a nuclear powered icebreaker. All vessels over 150 feet probably will be designed for work of some degree in ice.

The possibility of a new hybrid design coming off the drawing boards exists. Patrol cutters and tenders perform similar duties. Today the tender can, and does, do all the jobs assigned to the patrol cutter, as well as doing its own special aids to navigation work. The one disadvantage that the tender suffers is that it is generally of slower speed than the patrol cutter.



U.S. Coast Guard Official Photo

Figure 11.

U.S. Coast Guard Official Photo

Figure 12.

Any hybrid design cutter should be able to work minor aids to navigation, perform some logistics work, have provisions included for towing and work in ice, have excellent seaworthiness, and moderate to good speed for assistance work. Armament should be limited to that which might actually be used in an extreme case of law enforcement work. Provisions should be made in the design for the installation of anti-submarine weapons and detection gear, when the need exists.

The major technical characteristics to be considered in the design of any Coast Guard Cutter in their order of priority appear to be: seaworthiness; speed and maneuverability; cruising radius as such, or capacity useful in logistics work; necessary deck equipment for aids to navigation and assistance work; habitability; ability to work in ice; provisions for required electronic equipment; and provisions for military readiness. The proper weighting of these characteristics will, of course, depend upon whether a hybrid duty ship is desired, or whether a ship for a special type of duty, such as a lightship, is required.

Unless the duties of the Coast Guard materially change, the foregoing characteristics should be appropriate for many years to come.

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APPENDIX I

PART A—COAST GUARD HISTORY AND ENGINEERING DEVELOPMENTS

CHRONOLOGY OF IMPORTANT DATES

1716—First lighthouse on continent built at entrance to Boston harbor by Province of Mass.

1789—Lighthouse Service established by Congress. This service takes over 12 lighthouses and 7 buoys from the states.

1790—Revenue Marine established by Congress. Name later changed to Revenue Cutter Service. Authorized 10

sailing cutters 50′ in length. This was first regularly organized armed maritime service of the government.
1799—Revenue cutters in quasi-war with France, and were in all subsequent wars.
1820—First lightship on station, in Chesapeake Bay.
1837—Congress authorized winter cruising of the Revenue Cutters for rescue work.
1838—Congress created the Steamboat Inspection Service. Name later changed to Bureau of Marine Inspection.
1840—First tender; sailing vessel.
1844—First steam side-wheel Revenue Cutter, *Legare*.
1848—First coastal life saving apparatus, on N.J. coast.
1850—Congress established present buoyage system.
1857—First steam tender, side-wheeler, *Shubrick*.
1871—Congress established Life Saving Service.
1882—First iron lightship on station.
1884—Congress established Bureau of Navigation. First lighted buoy in use.
1891—First lightships with their own propulsion, other than sail.
1897—Revenue Cutter *Manning;* triple expansion engine; 205′; 16K; cutters essentially same for next 20 years.
1912—"A Night To Remember"; *Titanic* sinks.
1913—International Conference for Safety of Life at Sea. U.S. Coast Guard to operate International Ice Patrol. Minimum standards set for passenger ships.
1915—Revenue Cutter Service and Life Saving Service merged, and names changed to U.S. Coast Guard.
1916—Congress authorizes 10 C.G. air stations.
1920—18th Amendment; C.G. v.s. bootleggers. Large expansion of C.G. law enforcement work.
1921—First regular radio beacon service. First turbo-electric drive on *Tampa* class; 240′; 1500 ton; 16.2K.
1926—First use of diesel on *Active* class; 251′; 291 ton; 13.2K. Later C.G. cutters listed in Appendix II.
1939—Lighthouse Service becomes part of Coast Guard.
1942—President transfers Bureau of Marine Inspection and Navigation (BMIN) from Department of Commerce to Coast Guard for duration of war.
1945—U.N. group, International Civil Aviation Organization (ICAO) established. Sets up ocean stations and requirements for Rescue Coordination Centers (RCC).
1946—U.S. designates Coast Guard to operate its ocean stations. C.G. to operate RCC for water areas, and Air Force to operate RCC for U.S. land areas.
1946—BMIN made a permanent part of Coast Guard.
1948—International Conference on Safety of Life at Sea. Minimum requirements set on inspections, surveys, new construction, investigations, and life saving equipment for vessels. International Rules of Road revised. Established Intergovernmental Maritime Consultative Organization (IMOC) under U.N. Charter. Coast Guard one of several agencies representing U.S. on IMCO.
1949—Congress codifies and enacts into law, Title 14, U.S. Code, dealing with duties of Coast Guard.
1950—Magnuson Act; start of Port Security Program in peacetime.

PART B—STATUTORY AUTHORITY FOR THE DUTIES OF THE COAST GUARD

Authority appears in the following titles, United States Code:

14—Coast Guard
16—Conservation
18—Crimes and Criminal Procedures
19—Customs Duties
33—Navigation and Navigable Waters
46—Shipping
50—War and National Defense

PART C—DUTIES OF THE COAST GUARD

A. LAW ENFORCEMENT

1. Prevention of smuggling.
2. Enforcement of navigation laws
   a. Rules of road.
   b. Anchorage regulations.
   c. Obstruction to aids and waterways.
   d. Oil Pollution Act.
3. Enforcement of Marine Inspection and Safety Laws.
4. Enforcement of Motorboat Laws.
5. Patrol of marine regattas, yacht races, and spectacles.
6. Transportation of Federal Court in Alaska (Bering Sea Patrol).
   a. C.G. officers can legally act as U.S. Commissioners and Marshalls in Alaska.
7. Protection of Marine Life (result of International Treaties)
   a. Fur seal and otter in North Pacific.
   b. North Pacific halibut.
   c. Gulf sponge fisheries.
   d. Whaling fisheries.
   e. Tuna fisheries.
   f. Northwest Atlantic fisheries.
8. Assist any federal agency, state, or local government
   a. State Department—Voice of America; Campeche Patrol
   b. Defense Department—Aids to Navigation.
   c. Fish and Wildlife Service—Conservation Laws.
   d. Treasury Department—Immigration and Customs.
   e. Local governments—time of disaster.
9. General apprehension of U.S. citizens for crimes on high seas or any criminal in U.S. jurisdictional waters.
10. Enforcement of dangerous cargo and port security laws.
11. Boarding vessels to check for federal law infractions.

B. SAFETY AT SEA

1. Search and Rescue
   a. Operation of rescue coordination centers.
   b. Operation of vessels and air stations.
2. International Patrols required by treaties
   a. International Ice Patrol.
   b. Ocean station program.

3. Cooperation with United Nations Groups
   a. ICAO—air.
   b. IMCO—surface vessels.
4. Merchant Marine Safety and Inspection
   a. Merchant ship construction and equipment standards.
   b. Merchant ship inspections and investigations.
   c. Licensing, shipments, discharge, and welfare of merchant seamen and officers.

C. AIDS TO NAVIGATION

1. Operate and maintain system of lights, buoys, other minor aids, tenders, lightships, and electronic aids to navigation.
2. Icebreaking in U.S. harbors.

D. MILITARY READINESS

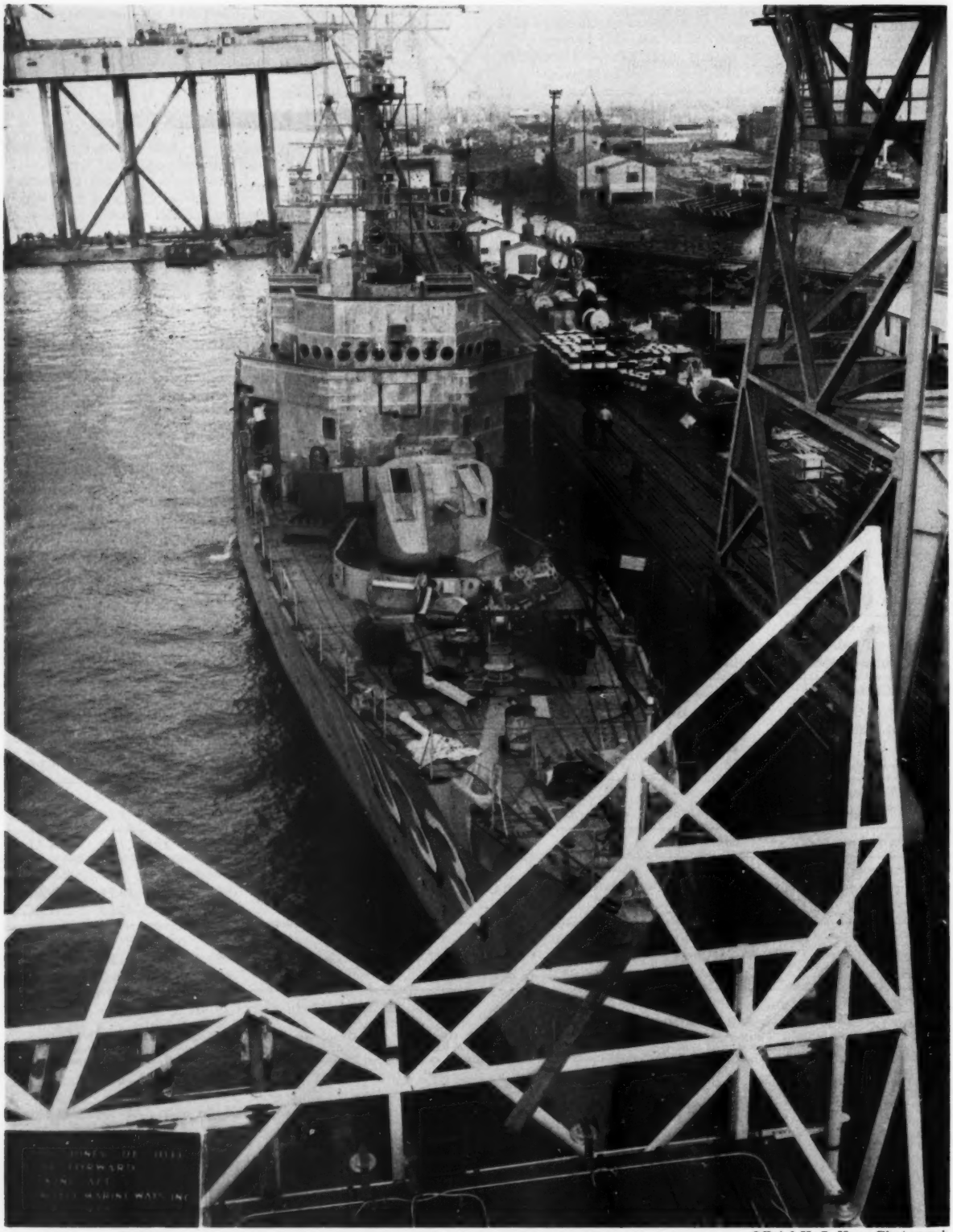
1. Operate as part of Navy when President directs.
2. Port security
   a. Prevent entry by sea of anything threatening security of U.S.
   b. Issue and check credentials for certain waterfront areas.
   c. Prescribe and enforce safety regulations for waterfront facilities, vessels, and anchorages especially with regard to handling of dangerous cargo.
   d. Complete control over all vessel movements when conditions warrant.
   e. Placing armed guards at key waterfront areas and coastal points when required.

APPENDIX II

TECHNICAL DATA—MAJOR U.S. COAST GUARD CUTTERS

	No.	LOA	BEAM	DRAFT	DISP.	GM	PROPULSION	SHP	SCREWS	SPEED	C.G. DESIG.
Cruising Cutter											
327′ Gunboat 1936**	6	327	41	15	2827	3.38	Geared Turbine	6200	2	19.0	WPG
Seaplane Tender, small	17	311	41	14	2800	3.10	Diesel Reduct.	5800	2	17.5	WAVP
255′ Gunboat 1942	13	255	43	17	1913	3.43	Turbo-electric	4000	1	17.5	WPG
Patrol Cutter											
165′ Patrol Craft, 1931	9	165	25	9	400	3.82	Diesel Direct	1300	2	13.0	WPC
125′ Pat. Cft., 1926	19	125	24	8	295	1.94	Diesel Direct	800	2	11.0	WSC
95′ "A" 1953	12	95	20	6	95	4.31	Diesel Reduct.	2200	2	19.0	WPB
83′ Pat. Boat, 1941	63	83	17	6	65	—	Various	1200	2	15.0	WPB
Buoy Tenders											
180′ "C" Class, 1944	20	180	37	13	1025	3.39	Diesel Elect.	1200	1	12.0	WAGL
*Lightships**	33	128	30	11	540	1.0	Diesel Direct	610	1	10.5	WAL
Icebreakers											
Wind Class, 1944	3	269	64	29	6515	7.57	Diesel Elect.	10000	2	15.0	WAGB
MACKINAW (Gt. Lakes)	1	290	74	19	5252	16.8	Diesel Elect.	10000	3	16.0	WAGB
Tugs											
Ocean, 1943	2	213	39	15	1940	1.83	Diesel Elect.	3000	2	13.0	WAT
110′ B Harbor	18	110	26	11	354	1.58	Diesel Elect.	1000	1	11.0	WAT

* This represents an average ship in this category.
NOTE: This is not a complete listing. Attempt was made only to list the major categories. In addition the Coast Guard operates 2 AK's, one for logistics of Pacific stations, and the other for Voice of America work in the Mediterranean. Also operated are one cable layer, one combination icebreaker-tender-logistics ship, miscellaneous buoy tenders, small buoy boats, harbor craft, surf boats, and the training bark EAGLE.
** Year indicated is completion for first ship of class.



Official U. S. Navy Photograph

Nearing completion the USS *Claud Jones* (DE 1033) is pictured alongside the builder's dock.

"THE MARINE ENGINEER AND NAVAL ARCHITECT"

WELDED FRAMES FOR MARINE DIESEL ENGINES

ACKNOWLEDGEMENT

Gunnar Hellström, of A. B. Götaverken, Gothenburg, Sweden, is the author of this article, which was published in the September 1958 and December 1958 editions of "The Marine Engineer and Naval Architect."

IN GENERAL machine and engine frames are constructed to such heavy scantlings that the average stresses are very low and even quite acute geometric irregularities cannot bring the stresses into the critical state. The growing need to obtain higher outputs from marine diesel engines has made it necessary to exploit the materials more efficiently in order to keep the different components within manageable proportions. Rising labor costs and, at times, shortage of material have increased the need for lighter construction and a considerable reduction in weight has been achieved by welding bedplates, frames and entablatures. There are several different types of welded-frame marine engines in use today.

In order to make them suitable for welded construction and welding materials, the designs have been modified from the original cast iron versions. Nevertheless, all types of welded engines have shown some tendency to crack and in some cases a complete bedplate has had to be replaced. All these failures are due to fatigue, if this term is used as a definition of the mode of failure and not as an explanation of the fundamental cause of breakage in service which is very often difficult to establish.

It can be shown, however, that when a crack occurs at any point, the stresses have exceeded the strength of the material at that particular point. The task must, therefore, be to diminish the peak stresses. Engine frames are stressed by external forces due to the movement of the ship, working stresses due to the gas forces, and internal forces, due to thermal stresses locked up when the plates were welded together. In case of fatigue the recurrent stresses caused by the gases are the most important ones.

DESIGN

The engines of cast iron construction were usually built in several parts, viz., bedplate, columns, distance-pieces and cylinder blocks, all secured together by long through-bolts which extended right from the top of the cylinder blocks to the bottom of the bedplate. After turning over to welded construction some manufacturers retained the stay bolt system but others dispensed with them and there has been much discussion as to whether stay bolts should or should not be used in welded engines.[1, 2] The forces in frames with and without stay bolts are compared schemetically in Figure 1. The frame with stay bolts has cross sectional areas $A_1+A_2=A$; $A_1=\frac{1}{n}A$. The frame without stay bolts has cross sectional area A. This gives constructions of the same total weight. For cast iron, where $E_1=2E_2$, the stresses obtained are:

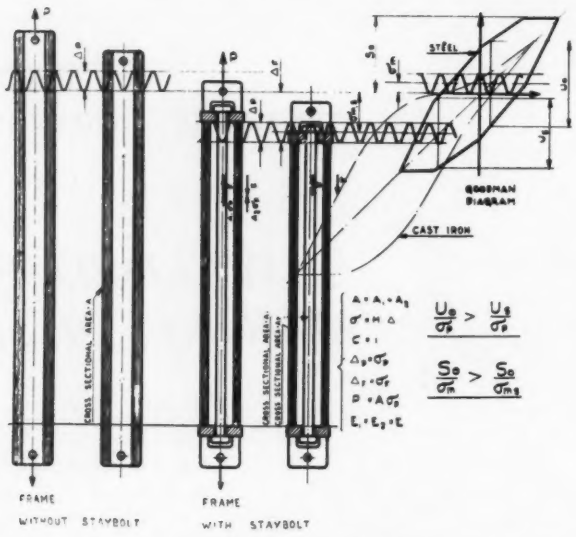


Figure 1. Forces in engine frames with and without stay bolts.

Frames with stay bolts

$$\sigma=-K\frac{P}{A}\pm\frac{n}{2(n+1)}\frac{P}{A}$$

Frames without stay bolts

$$\sigma=\frac{P}{2A}\pm\frac{P}{2A}$$

Cast iron can withstand compressive stresses of considerably higher magnitude than the tensile stresses shown in the Goodman diagram. A really high numerical value of F or K does not harm the frame. The notch sensitivity of cast iron is really very low and there is no danger of severe stress concentrations. By using stay bolts in cast iron frames it is possible to exploit the special strength of the material further and to decrease the engine weight. For steel, where $E_1=E_2$, the stresses obtained are: Frames with stay bolts

$$\sigma=-K\frac{P}{A}\pm\frac{P}{2A}$$

The smallest value of F that can be chosen to avoid leakage is

$$F=P\frac{A_2E_2}{A_1E_1+A_2E_2}$$

Actually A_2 is a function of l

Thus, when $E_1=E_2$, $F=\dfrac{P\,A_2m}{A_1+A_2m}$

where $A_2m=\dfrac{1}{l}\int_0^l A_2dl$

If $A_2m=5A_1$ and the smallest cross sectional area $A_2=A_1$

$$\sigma_F=-\frac{F}{A_1}=-\frac{P}{A_1}\frac{5}{6}=-\frac{5P}{3A};$$

$$\sigma_P=\pm\frac{1}{2}\frac{P}{A}$$

$$\sigma=-\frac{7}{6}\frac{P}{A}\pm\frac{P}{2A}$$

and thus $K=\dfrac{7}{6}$

Frames without stay bolts

$$\sigma=\frac{P}{2A}\pm\frac{P}{2A}$$

In practice it is impossible to obtain the smallest value of F, and thus K often has a much higher value. With increasing K the factors of safety decrease and these are specially accentuated at the areas of least cross section.

Severe stress concentration is likely to be met at the stay-bolt landings and the belief that the stay-bolts will protect the frames and bedplates from severe stresses is not in accordance with facts. In case of a welded diesel engine the design without stay bolts is the most suitable having regard to the resistance of the material to the stresses arising from the cylinder pressure and to the avoidance of features making for stress concentrations. Absence of the stay bolts makes the design of a relatively simple engine frame possible, Figure 2, with the possibility of fabrication in larger units and the elimination of a number of machined and bolted joint faces. The Götaverken design is an excellent example of this type. The bedplate containing the lower half of the crank case consists of a suitable

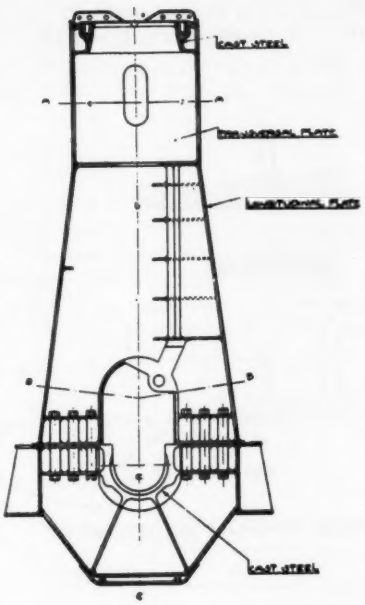


Figure 2. Early engine frame constructed without stay bolts.

number of sections fitted together with bolts. Each section consists of transverse girders, one between every crank, built in between two longitudinal girders. The upper half of the crank case and the block for each cylinder are formed into a box-type entablature with a heavy steel casting welded into the top. This casting is provided with heavy fitted bolts which thus connect all the entablatures enabling them to share the loads imposed by the combustion gas forces. These castings also carry the long cylinder head holding-down studs. Heavy bolted connections are used to secure the entablatures to the bedplate.

Elementary methods may be used for calculating the average stresses due to external forces in the bedplates and entablatures. In the case of the design without stay bolts, only the gas forces based on the indicator diagram need be considered. The method of calculation has been published.[7] The average permissible stresses will depend on the shape of the single elements and their stress raising effects. A prime requirement when designing for fatigue conditions is to avoid the occurrence of stress concentrations. The transverse girder in the bedplate shown is an example of a well-designed element. The bearing housing is made of cast steel, smoothly rounded and formed with extension webs for butt welding to the plates of the transverse girder. The bolts used between the entablatures and the bedplate are of considerable length. Not a single crack has been found in a bedplate of this design, the earliest of which was delivered in 1945, evidence that a properly designed and manufactured welded component can easily meet the requirements of heavy duty in marine service.

Some cracks have been experienced in the entablatures of this engine type, Figures 3 and 4. The most severe ones have occurred at the opening for the scavenging air. This opening disturbs the stress flow in a similar manner as does a circular hole. The bolt holes, which are located in the close vicinity of the opening, increase the concentration of the stresses. As seen in Figure 5 the peak stress is more than six times the average stress, and over 10 kg/mm^2 has been obtained by measuring with strain-gauges within a bolt hole.[3] The problem of the stress flow around an opening of the shape here used cannot be solved exactly, as only for circular and elliptical holes are there exact solutions. Therefore, the result obtained for a circular hole is used. The difference is of no importance, and the accuracy is sufficient for practical purposes. To avoid this stress concentration the bolt holes were removed from the vicinity of the opening and the edge was supplied with a symmetrically located reinforcement. Again, the value of reinforcements at openings cannot be calculated by exact methods. It is

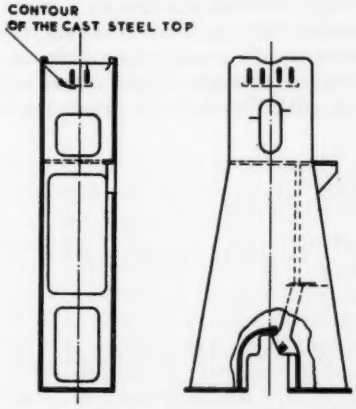


Figure 3. Cracks at openings for scavenge air.

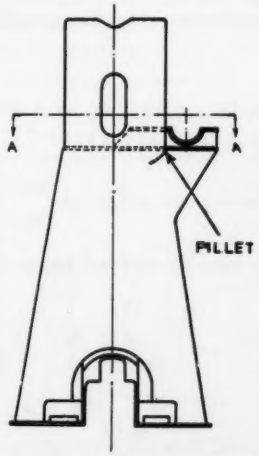


Figure 4. Fillet at camshaft entablature.

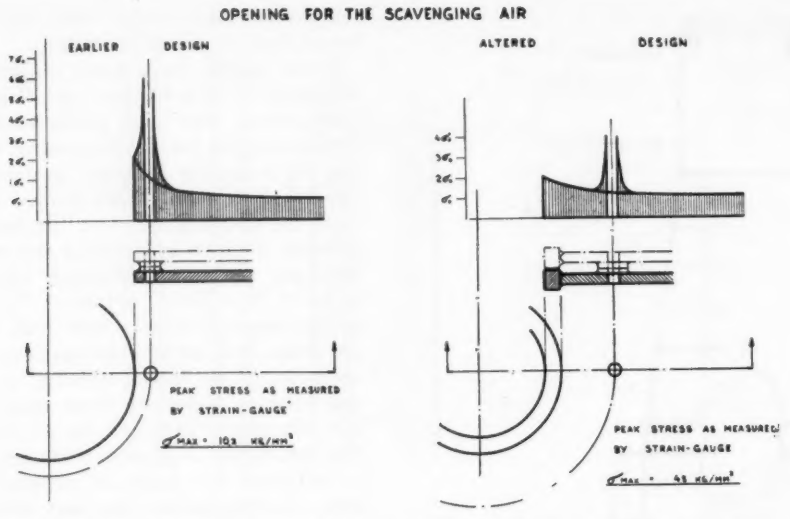

OPENING FOR THE SCAVENGING AIR
EARLIER DESIGN
ALTERED DESIGN
PEAK STRESS AS MEASURED BY STRAIN-GAUGE
PEAK STRESS AS MEASURED BY STRAIN-GAUGE

Figure 5. Showing much reduced peak stress after modifying design of opening for scavenge air.

supposed that the problem can be equalized by a round disc on its outer edge loaded with tensile stresses.[4, 5] The elementary theory for a heavily curved beam can then be used to obtain a value for the reinforcement of the circular hole in a plate loaded by tensile stresses. Thus, when indications in accordance with Figure 6 are used, the relations are:

$$N_o = \sigma_o t k a$$
$$t\sigma_o d b = N_o \sin\phi d\phi$$

$$\uparrow \; -N_o + \frac{N}{\cos\phi} + \sigma_o t k a (1-\cos\phi) = 0; \quad N = N_o \cos^2\phi$$

$$\overset{>}{\underset{o}{\frown}} \; M - M_o - N_o r + N r + \int_o^\phi N_o \sin\phi \cos\phi k \,.\, a d\phi = O$$

$$M = M_o + N_o r - N_o r \cos^2\phi - N_o k a \tfrac{1}{2} \sin^2\phi$$

Further the equations must be satisfied:

$$\int_o^{\frac{\pi}{2}} M r d\phi = O$$

and the expressions are obtained $\begin{cases} N_o = \sigma_o t k a \\ M_o = N_o (\frac{1}{4} k a - \frac{1}{2} r) \end{cases}$

The stress at a heavily-curved beam is:

$$\sigma = \frac{N + \dfrac{M}{r}}{A} + \frac{M}{J} \frac{Z}{1 + \dfrac{Z}{r}}$$

when $J = -r^2 A + \Sigma r^2 bi \, ln \, \dfrac{\rho i + 1}{\rho i}$

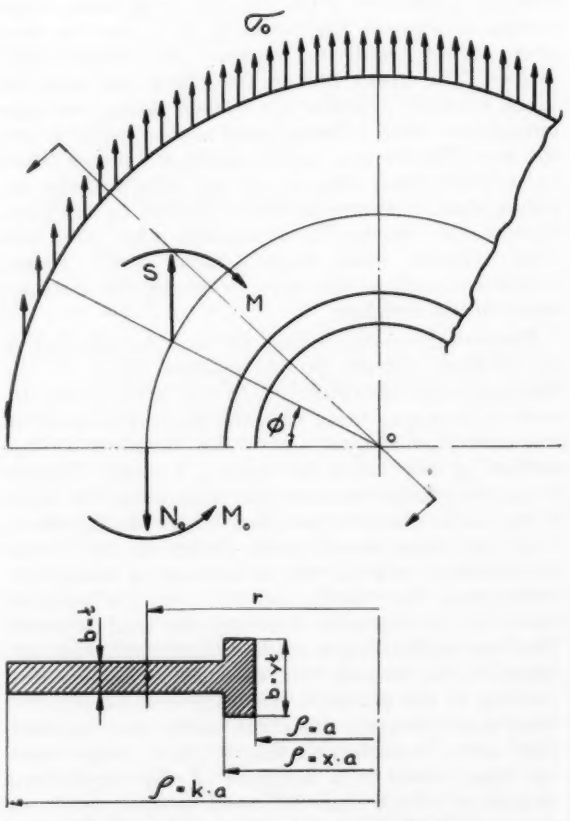


Figure 6. Stresses in a heavily curved beam.

For a point on the line A-A at a distance s a from the central point the stress becomes

$$\sigma=\frac{\sigma_0 k}{2Q}\left[\frac{Q+kU}{U}+\frac{1}{s}\,\frac{(kU-Q)(2sU-Q)}{Qln\,(kx^{y-1})-2U^2}\right]\ldots\ldots A$$

where

$U=k-x+y(x-1)$

$Q=k^2-x^2+y(x^2-1)$

When s=1 the peak stress at the edge of the hole is obtained. Then, when y=x=1, σ must be $3\sigma_0$ for an agreement with the exact theory. When k=3.3 this value is obtained which is used in the continuation. By imposing different values of x and y in the table, Figure 7 is obtained. It appears that the most suitable values of δ/a are to be found between 0.15 and 0.40. A smaller value will give insufficient reinforcement and a higher value will not give reinforcement corresponding to the weight inserted. The most suitable values of H/t are between 2 and 3.5 for the same reasons. The actual reinforcement was estimated in accordance with this result and the stress flow has been calculated by using the formula A. By measuring with a strain gauge in the bolt hole a stress of the magnitude 4.4 kg/mm² was measured. A decrease of 50 per cent has been obtained. It is often argued that a better result would be obtained by an elliptical opening.[6] Obviously this is in accordance with real facts if the opening can be made longer and narrower in accordance with the formula

$$\sigma=\sigma_0\left(1+2\,\frac{a}{b}\right)$$

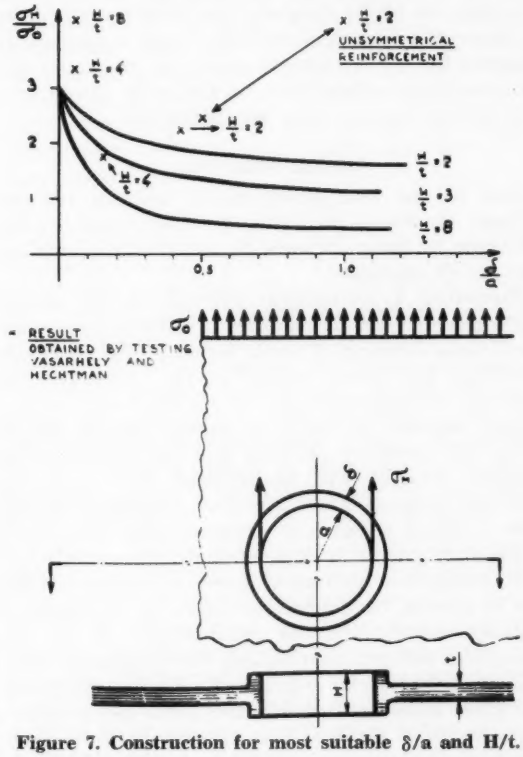


Figure 7. Construction for most suitable δ/a and H/t.

But in practice the area of the opening must be constant and an increase of the length is impossible due to the vicinity of the cast steel top and the intermediate plate. To determine the influence of the shape an optical stress analysis was made by Chalmers Technical University. The result, given in Figure 8, gives no reason for altering the shape of the opening. The openings on the front and rear of the entablatures have been shaped partly in agreement with the related results. At the circular opening in front there are holes with threads cut in the reinforcement for studs. Screwing the studs in prevents the free deformation of the holes and prevents the rise of severe stresses when the entablature is acted upon by the combustion forces. The lower joints of the cast steel top were formerly made by abrupt welds and cracks were experienced near the ends of the welds, where a detrimental stress flow occurred due to the sudden change of the material thickness. Now the joints are continued around the whole cast steel top, and the failures by cracking have ceased. The upper main bearing shells were earlier held by compressive bolts. These are no longer used, resulting in a slight increase of the average stress in the lower part of the entablature but elimination of the severe disturbance of the stress flow which the attachments for these bolts caused. The beams supporting the guide plates have been lengthened and supplied with smoothly formed ends.

The maneuvering system is arranged within the intermediate entablature. Formerly this was connected to the neighboring entablatures but, due to the temperature difference, the maneuvering entablature was subject to a considerable static load. Cracking also occurred in the fillet for the step for the cam shaft drive, where a slight stress concentration also was found. Now the intermediate entablature is disconnected from and independent of the contiguous entablatures and thus relieved from the gas forces. The cracks here mentioned have been the most common ones. To get an even stress flow the cast steel top has been reinforced to prevent deformation by bending.

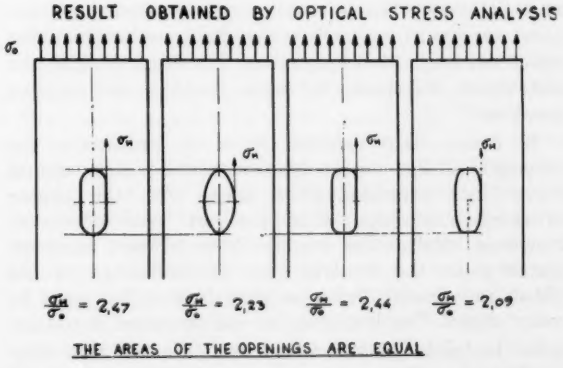


Figure 8. Different suggestions for scavenge air opening.

The welded joints used are entirely designed as continuous welds. The main bearing housing in the bedplate and the reinforcements of the openings in the entablatures are designed for complete fusion welds with a full bevel to ensure full penetration. But fillet welds are used generally at cross joints without preparation of the edges of the plate. The joints between the cast steel top and the plates in the entablatures are of lap type with fillet welds. Of course butt welds and welds with complete fusion are better than those used. However, the lengths of the joints are considerable and the stresses are small and cause no trouble. The fitting is also easier. The cracks obtained have never started in the welds and it is Götaverken's experience that, while the surfaces of the welds must be smooth, included cavities such as those caused by an incomplete fusion did not cause trouble at reasonable magnitude of the stresses. However, if all geometrical disturbances caused by holes, grooves, etc. are avoided, the magnitude of stresses is limited by the welds used. To obtain a decrease of weight it may be advisable to switch to complete fusion welding. However, the design must then be carefully overhauled to avoid intersection of different joints, raising the stresses seriously; e.g., if the joints at the cast steel top are designed as butt welds, it may be necessary to curve the corners of the entablatures to avoid dangerous stress concentrations.

STRESSES DUE TO SHIP MOTION

It has often been said that a stiff bedplate can relieve the engine from stresses due to the movements of the ship but this does not agree with the facts. In a vessel with the machinery amidships clearly the engine room double bottom can be deflected due to hydrostatic pressure. The moment of inertia of a transverse section of the bedplate is only a fraction of that of the double bottom, and it is therefore impossible to stiffen up the double bottom by increasing the bedplate scantlings. If the load of the ship is distributed so that most of the weight occurs at the ends, there will be hogging deformation and stresses, while loading with the weight amidships will cause sagging deformation and stresses. The stresses and deformations are most severe in seas when the distance between the wave crests is about equal to the ship's length and the vessel alternates between hogging and sagging positions.

If these deformations were to influence the strength of the engine frames, more cracks should have been recorded from ships with the engine situated amidships. This has not been observed, however, and as the engines have always been situated near the neutral line the influence of the ship's movements from sagging to hogging must be very slight. The best results are obtained if the engine bedplate is permitted to slide on the shipbuilder's seatings.

In order to estimate the influence of ship movements in a heavy sea, measurements were taken on board a bulk ore-carrier. At that time, the equipment available for dynamic measurements was not sensitive enough and so these measurements were made when loading and unloading and heeling the ship. Strain gauges were placed on the upper and lower flanges of the longitudinal bedplate girder and at different points on the entablatures. The draft variation was 7 ft. and a list of 15 degrees was applied but only very small values were obtained. As the center of gravity of the engine is lower than the roll center of the ship, the dynamic forces may well diminish the forces acting on the engine frame. By using an M/EJ diagram the deformation of the bedplate can be calculated. By using a Bon Jeans diagram the peak moment could be estimated and the maximum stress due to the ship movements $\sigma_s = \pm 2.5$ kg. per sq. mm. was obtained.

STRESS RELIEVING

Due to the heat involved in welding residual stresses remain in the parts so joined together. The influence of these stresses on fatigue behavior has been much discussed.[8]

Numerous investigations carried out on welded joints have shown no significant difference between the fatigue strength of joints tested in the as-welded condition and those tested in the stress-relieved condition. It is argued that where there are residual tensile stresses initially, a super-imposed tensile stress will produce yielding and on unloading the residual stresses will be reduced. The reduction may be complete after the first loading cycle and even residual stresses of reversed sign may result if the load applied is large enough. Where this is in fact an effective method of stress relieving, in practice it cannot be applied as there is no means of suitably overloading the entablatures. If the entablatures are not overloaded, the residual stresses will nevertheless be reduced by the gas forces but the peak stress will touch the yield point during every cycle in service. This lies on the limit of the Goodman diagram at 2×10^6 stress cycles. Thus, the factor of security obtained is unity and it would be hazardous to use the construction beyond 2×10^6 stress cycles. While the stresses due to the ship movement at sea may reduce the peak stress if yielding occurs, it is impossible to discount the influence of these overloadings due to the highly variable amplitudes and frequencies. Further the manufacturers of a certain type of machine have experienced that unrelieved frames cannot be relied upon to give satisfactory work,[9] the residual stresses having been relieved in time with consequent deformation. Obviously this might happen in the case under discussion when due to the deformation alteration of load distribution occurs which could cause overstressing of certain elements. Measurements in accordance with Dr. Gunnerts' well-known method have been carried out. Residual

stresses of 12 kg. per sq. mm. have been found in unrelieved entablatures and of 25 kg. per sq. mm. in unrelieved transverse bedplate girder assemblies.

Götaverken practice always has been to stress relieve by heat treating only the transverse girder assemblies before they are welded to the longitudinal girders. This practice is justified by the fact that strain gauge measurements of the stresses in the bolts securing the entablatures to the bedplate have clearly shown that only the inner bolts are subjected to cylinder loading pressure. Consequently the welds between the transverse girders and the longitudinal girders are not subjected to loading from the gas forces. The earlier engines were delivered with unrelieved entablatures. Nowadays, stress relieving is done by heat treating of the complete entablatures after welding. In order to discover stray cracks due to the welding and material defects every entablature and bedplate is subject to Magna-Flux examination. Ultrasonic and X-ray tests are also used.

MATERIALS

The first welded engines were made of ordinary shipbuilding quality plate. Nowadays, materials corresponding to the requirements for P 5 and NV W are mostly used. For the entablatures of Götaverken engines a steel of higher quality—2110 S—is used. The carbon content is 0.15 per cent and the manganese content is 1.5 per cent. Further it is killed and treated in order to give a fine grain structure. The ultimate tensile stress is about 52 kg. per sq. mm. It is chosen in order to give a high endurance limit, good weldability and security against brittle failure.

Fatigue testing of unnotched test pieces by repeated loading (from nearly zero to maximum load) has shown that the maximum stress or $2\sigma_a$ by $N = 2 \times 10^6$ stress cycles is equal to the yield point. Further, it is observed that the fatigue cracks start at points where yielding occurs. In view of these facts it may be concluded that a material can endure at most 2×10^6 stress cycles when the yield point is touched. However, the fatigue testing of notched pieces at 2×10^6 stress cycles reveals a relatively higher notch sensitivity for high quality steel than for common ship plate. One may ask "How can notch sensitivity be defined?" Parts of an engine structure must sustain 10^6 stress cycles *per week* and up to $1{,}000 \times 10^6$ stress cycles during a normal life. Can this case be represented by a test at 2×10^6 stress cycles? The geometrical form factor α is the relation between the maximum peak stress and the average stress in a stress flow disturbed by a notch in a completely homogeneous and elastical material. All materials consist of parts, crystals, grains, etc. The distribution of the stress depends on the size of these parts. In a small area at the bottom of the notch an average stress is obtained. Therefore, in a real material the maximum peak stress $\sigma_{max.} = \beta' \sigma_{ave.}$ is obtained instead of $\sigma_{max.} = \alpha \sigma_{ave.}$ β' is called notch sensitivity and is consequently smaller than α. On unloading no residual stresses remain. Neuber has published tables and formulae for obtaining α and β' based on tests at static loadings.[10] However, the notch sensitivity is mostly used in calculating fatigue resistance.

When a notched test piece is sufficiently stressed, yielding occurs, resulting in a plastic stress flow. After unloading, a proportion of the residual stresses remain. Usually the notch-sensitivity β is determined as the relation between the average limit stresses obtained by fatigue testing at 2×10^6 stress cycles of notched and unnotched test pieces. Here a material with a low yield point (low ultimate tensile strength, low quality) will give a low notch sensitivity value. By testing similarly notched test pieces of different materials very small differences of endurance limit are obtained.[12] However, the problem with the residual stresses here is similar to that previously discussed regarding the residual stresses from welding. Frequently low yield point material gives a better result in fabrications which have to sustain only a limited number of stress cycles of varying amplitudes and in consequence with the above it is of little value to stress relieve these constructions by heat treating.

Unfortunately, this experience cannot be applied to the design of an engine frame or entablature which is likely to endure 2×10^6 stress cycles in a fortnight. The gas forces produce cyclic loads with tolerances smaller than any fatigue testing machine can offer. Obviously, overloading may occur due to ship movements, but the stresses arising from these movements being independent of the repetitive gas forces and their frequency and amplitude vary widely. The maximum stresses must therefore be very low and should not result in yielding at any point of the construction exposed to the gas forces. A high-quality steel with a high U.T.S. is therefore to be preferred.

The relations between α, β', β are shown in Figure 9. The load P is the same but its effect on four different materials is shown. The first is completely plastic, the second and third are completely elastic. In the third, however, the size of the grains influences the stress flow. The fourth material has residual stresses after unloading.

Tests to estimate the elastic limits are being carried out in order to obtain further information about the elastic properties of the materials. The elastic limit is defined to be the highest stress to which the material can be forced, when the residual strain is lower than 10^{-4} per cent. The experimental work is supervised by Dr. B. Kjerrman, former chief of the central laboratory of the A.B./Svenska Kullagerfabriken. As mentioned previously, a material with a high U.T.S. always gives a better result than that obtained with ordinary plate when fatigue testing unnotched test pieces. It is therefore essential to design the structure as free as possible from stress

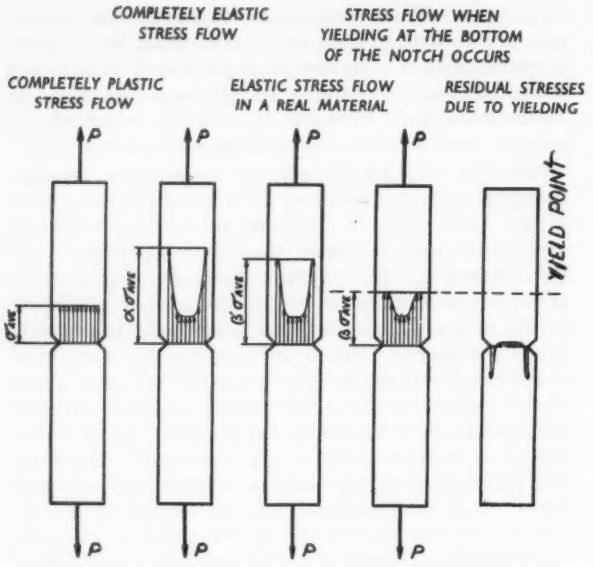

COMPLETELY ELASTIC STRESS FLOW
STRESS FLOW WHEN YIELDING AT THE BOTTOM OF THE NOTCH OCCURS
COMPLETELY PLASTIC STRESS FLOW
ELASTIC STRESS FLOW IN A REAL MATERIAL
RESIDUAL STRESSES DUE TO YIELDING
YIELD POINT
P

Figure 9. Stress flow diagrams showing relationship between α, β' and β.

concentrations in order to exploit the properties of high-quality material. A high U.T.S. usually is obtained from a high carbon content. But the tendency of steel to crack when welded increases with the increasing alloy content, particularly with a high percentage of carbon.[11] Carbon is, therefore, partly substituted by manganese, but even this alloy has an upper limit with regard to the weldability. It is argued that materials with good weldability and of much higher U.T.S. than mentioned here should be used, but such materials are not available in dimensions suitable for the actual purpose. Brittle fractures do not seem to have caused the failures mentioned here. Rigorous claims to impact resistance are of no value. It should be mentioned that corrosion fatigue may be present and experience has shown that it is of some value if the steel is killed.

SUMMARY

A welded engine frame, without stay bolts, is to be preferred in order to make the best use of the material. Such a layout, without stay bolts, makes the overall design relatively simple, with the least number of details, as shown by the Götaverken engine. By heat treating the entablatures and the transverse girders of the bedplate, the residual stresses in parts exposed to the gas forces are relieved.

When calculating the average stresses in these parts alone the external gas forces as obtained by the indicator diagram need be considered, as ship movements give rise to insignificant stresses. Measures must be taken to avoid stress concentrations in order to resist fatigue stresses. The Götaverken bedplate is an outstanding example of this practice for not a single crack has been discovered in the bedplates of any of the 220 welded marine diesel engines hitherto delivered by Götaverken or its licensees.

Some cracks have been experienced in the entablatures. By measurement and calculations the alternating peak stress $\sigma = \pm 5$ kg. per sq. mm. was obtained where the most severe cracks had occurred. Combined with the residual stresses in unrelieved entablatures, the state of stress may result in a factor of security as low as unity. By altering the design an alternating peak stress $\sigma = \pm 2.2$ kg. per sq. mm. was obtained at the corresponding point. By heat treating of the entablatures the residual stresses have been relieved. If the influence of the ship's movements is considered the resulting stress may be $\sigma = 2.2 \pm 4.7$ kg. per sq. mm.

In bedplates a material corresponding to the requirements of Lloyd's Register's P 5 and Norske Veritas W NV is used. For entablatures a material 2110 S with the carbon content 0.15 per cent is used. The steel is killed and fine grain treated. By this material an increase in the limited stresses of the Goodman diagram is obtained.

The measures taken have resulted in the existence of a real safety factor as is shown in Figure 10. This statement is proved by the experiences in service. All engines delivered since 1954 have been completely free from cracks.

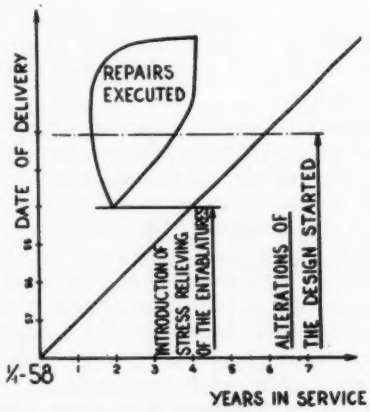


Figure 10. Diagram showing casualty and service experience with Götaverken welded engines.

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The construction of the nuclear ship SAVANNAH is proceeding close to its original schedule. Due to be launched this summer, the ship is now receiving the reactor components. Two steam generators have been shipped from the Babcock and Wilcox plant in Barberton, Ohio, and the 100-ton pressure vessel was scheduled for shipment in April.

Every reasonable safety precaution has been taken in the design of the ship. Extra shielding is provided, the reactor is located where it would be least likely to suffer collision damage, and a 168-point monitoring system is provided to indicate any operational irregularity.

A departure from usual commercial practice is the installation of planetary reduction gears, said to be 70 per cent lighter and to achieve about 70 per cent savings in space, as compared to a conventional reduction gear.

A proposal has been set forth for the construction of a graphite moderated, gas cooled reactor marine propulsion plant using helium as the coolant. The coolant, emerging at 950°F, would be used to generate superheated steam, adaptable for use with conventional steam propulsion equipment.

It has been proposed jointly by Ford Instrument Company, Maryland Shipbuilding and Dry Dock Company and Isbrandtsen Company, that the installation be made in a cargo-tank section of a ship. After a full testing program, the ship section would be installed in the SS HANS ISBRANDTSEN a 33,000-ton tanker launched in 1957.

—from MECHANICAL ENGINEERING, 1959

A recent report by the Committee on Oceanography of the National Academy of Sciences has emphasized the importance of an expanded program in oceanography. The report recommended that the government expand its support of marine sciences so as to at least double the current basic research activity within ten years. To achieve this and other objectives, it was specifically recommended that 70 research ships of 500 to 2200 tons displacement be constructed by 1970; that manned submersible devices that can operate on the bottom of most oceans be developed; that the disposal in the ocean of radio-active wastes be regulated and monitored; and that international cooperation in oceanographic research be achieved. It was stated that action on at least the recommended scale was necessary to prevent jeopardizing the position of this country in oceanographic science with respect to other nations, thereby accentuating serious military and political dangers.

—from SCIENCE NEWS LETTER
February 28, 1959

The most recent and most powerful of the Navy's large aircraft carriers, USS INDEPENDENCE (CVA-62), was commissioned on January 10 this year in the New York Naval Shipyard. Measuring 1046 feet long and 252 feet wide, her flight deck has an area of more than 4 acres. The flight deck is equipped with 4 steam catapults and 4 elevators which permit rapid and efficient handling and launching of airplanes having loaded weights of over 70,000 pounds. Her four shafts are powered by steam from eight oil-fired boilers and drive twenty-two foot propellers. The ship carries two 30-ton anchors. Each link of the anchor chain weighs 360 pounds. A 400 line telephone exchange, a 1050 ton air conditioning plant, 2 bakeries, 6 galleys, and an 84 bed hospital are among the services provided for her more than 4000 man crew.

—from BUREAU OF SHIPS JOURNAL, April, 1959

N. H. JASPER, Dr. Eng., and R. L. BROOKS, CDR, USN

STATISTICAL PRESENTATION OF THE MOTIONS, HULL BENDING MOMENTS, AND SLAMMING PRESSURES FOR SHIPS OF THE AVP TYPE

THE AUTHORS

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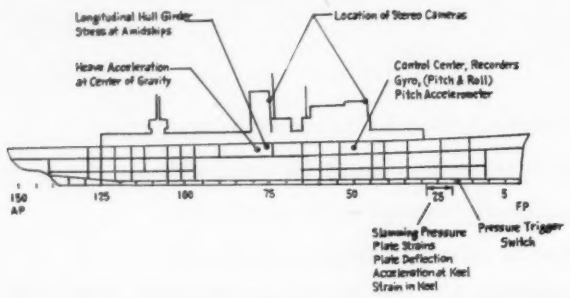
Commander Brooks is an Engineering Duty Officer who has served 7 years in the United States Fleet, 4 years in shipyards and 5 years in research and technical work at the David Taylor Model Basin and the Bureau of Ships. He is presently Senior Assistant to the Supervisor of Shipbuilding and Naval Inspector of Ordnance, Camden, New Jersey.

INTRODUCTION

THE Bureau of Ships is sponsoring a long-range investigation of strains in ships at sea for the purpose of evaluating and improving methods for the design of the ship girder and its structural components. Instrumentation has been developed and installed on various types of ships to collect information on the wave loads, streses, and motions which ships experience in service. During the wintetr of 1954 and 1955, measurements were carried out on the USCG *Unimak* (formerly AVP31) during operation in the North Atlantic Ocean. One of the main objectives of this work is the collection of sufficient data on ship motions and longitudinal hull girder stresses to determine, by statistical methods, the frequency distributions of these quantities for different combinations of sea conditions, ship speed, and ship heading relative to the waves. For a complete background and general discussion of these trials, see item 1 in the list of references at the end of this paper .

This report presents the distributions of the motions and bending moments* to be utilized for de-

* The hull bending moments due to flexure in the longitudinal plane of the ship were deduced from the strain measurements and the section modulus applicable to the strain-gage location.



Midship Section Modulus (for location of strain gages) 11,000 ft.-in.^{2}	
Midship Section Moment of Inertia	761 ft^4
Block Coefficient	0.571
Midship Section Area Coefficient	0.972
Prismatic Coefficient	0.588
Waterplane Area Coefficient	0.703

Figure 1. Profile and Characteristics of USCGC *Unimak* (AVP10-Class Vessel).

sign purposes. To devise these distributions, it is necessary to specify the ship operations for which the vessel is to be designed. The term "mission" will be used here to define the ship's assigned operational pattern. One component of this mission is the aggregate of sea conditions under which the vessel must operate. It will be assumed that the ship will operate in the North Atlantic Ocean inasmuch as this probably represents more severe sea conditions than the vessel will normally experience and thus is on the safe side.

The sea conditions will be specified in terms of a significant wave height.* Estimates of the significant wave heights are made by weather observers stationed on a number of weather ships at various locations in the Atlantic Ocean. These observations have been made at 3-hr. intervals since 1947. It has been found that the frequency distribution of these significant wave heights is approximately logarithmically normal.** The Weather Bureau's records of significant wave heights have been utilized here to evaluate the sea conditions to be expected in the North Atlantic Ocean.

During the at-sea phases, oscillographic recordings were made of actual variations of roll and pitch angle, heave accelerations, and hull strains as the ship responded to wave-induced loads. In general, ½-hr. continuous records were taken for each combination of ship speed, heading, and sea condition. Typical oscillograms are shown in Figure 9. Instruments were located as shown in Figure 1.

The pressures incident to slamming acting on the ship's bottom were measured by seven pressure gages installed on the *Unimag*.[5] Similar but more limited data were obtained during trials[6] of a sister ship, the USCGC *Casco*.

* The significant wave height was obtained by averaging the observed highest wave in each of a number of groups of waves. Note that the term "significant height" as used here is not synonymous with the statistical meaning of "significant" value which is defined as the average of the upper third highest values.
** See item 2 in the list of references.

STATISTICAL BACKGROUND

The wave heights, ship motions, and hull bending moments experienced under a given set of conditions will be described or specified in terms of their distribution patterns or, mathematically speaking, their distribution functions.

For illustrative purposes, consider one of the variables, for example, wave height. All wave heights are considered to be members of a statistical "population." The distribution function (d.f.) of wave heights indicates the relative probability $p(x)$ of encountering a wave of a given height as a function of that height. Figure 2a illustrates this distribution function. (Similar illustrations are given for the ship motions in Figures 3 and 4.) The area under the curve up to a value x_i is the integral of the d.f. up to the value $x=x_i$; it is equal to the function of all members of the population of wave heights which have a height less than x_i. Mathematically

$$P(x)=\int_0^x p\,dx \text{ and } P(x\to\infty)=\int_0^\infty p\,dx=1. \quad [1]$$

P is a function of x, and this function is designated as the cumulative distribution function (c.d.f.) of x. $P(x)$ is numerically equal to the probability that a value chosen at random from the population is less than x.

A discussion of the statistical methods utilized here is given in References 2 and 7.

There is considerable evidence[2] to indicate that the distribution of wave heights corresponding to any *one* given sea condition is of the one-parameter type known as the Rayleigh distribution which is defined as

$$P(x)=1-e^{-x^2/E}$$

where E is independent of x. Thus the probability is defined by a single number* E. On the other hand, when the heights of all waves experienced over a long period of time, say over several years, are considered, then the evidence indicates that the logarithm of the wave height is approximately normally distributed, that is, the two-parameter log-normal distribution describes the situation. The log-normal distribution is defined as follows:

$$p(\log x)\,d(\log x)=\frac{1}{\sigma\sqrt{2\pi}}\,e^{-\frac{(\log x-\mu)^2}{2\sigma^2}}\,d(\log x)$$

where μ is the mean value of $\log x$ and σ is the standard deviation of $\log x$.

Reference 2 shows that these two types of distribution also describe the response of the ship to the

* E is the mean value of x^2.

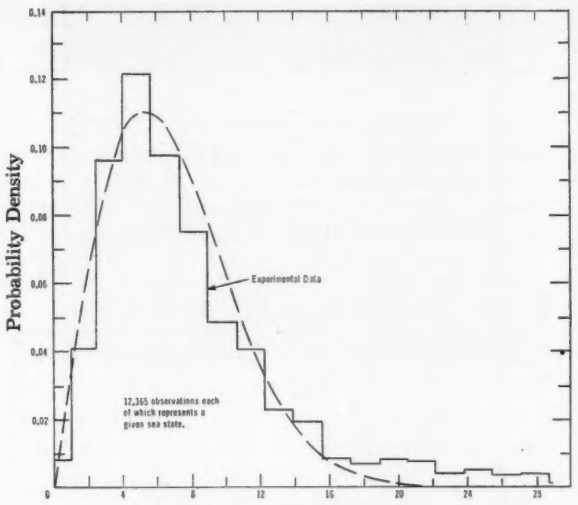

Probability Density
Experimental Data
12,365 observations each of which represents a given sea state.
Significant Wave Height, feet

Figure 2a. Distribution Function.

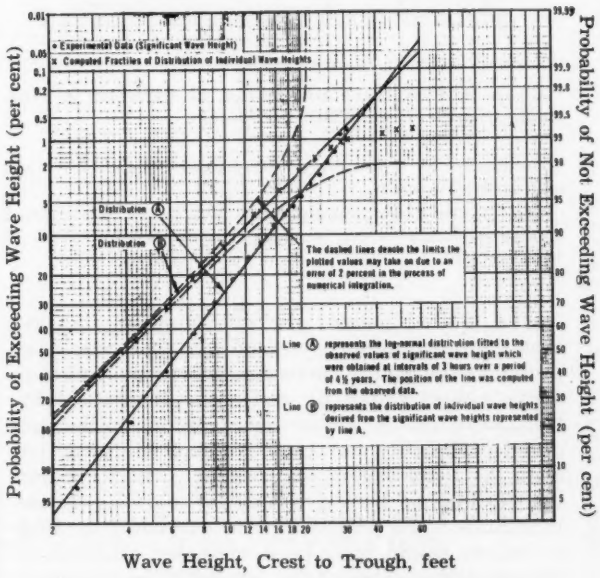

Experimental Data (Significant Wave Height)
Computed Fractiles of Distribution of Individual Wave Heights
Distribution A
Distribution B
The dashed lines denote the limits the plotted values may take on due to an error of 2 percent in the process of numerical integration.
Line A represents the log-normal distribution fitted to the observed values of significant wave height which were obtained at intervals of 3 hours over a period of 4½ years. The position of the line was computed from the observed data.
Line B represents the distribution of individual wave heights derived from the significant wave heights represented by line A.
Probability of Exceeding Wave Height (per cent)
Probability of Not Exceeding Wave Height (per cent)
Wave Height, Crest to Trough, feet

Figure 2b. Cumulative Distribution Function.

Figure 2. Distribution of Heights of Ocean Waves at Weather Station C, 52° N 37° W, North Atlantic Ocean.

This distribution is based on 12,365 observations made over a period of 4½ years by U.S. Weather Bureau personnel.

waves. For the sake of brevity, the distribution applicable to homogeneous conditions of the sea, ship speed, and course will be called "short-term" distribution, whereas the function which represents the distribution when the seas, ship speeds, and courses are allowed to vary over a range of conditions, will be designated as "long-term" distributions.

The distribution pattern will, at a glance, give the probability of exceeding any given magnitude of motion or stress. It also can be applied to the prediction of the largest magnitude to be expected in a given number of variations. For application to design for endurance strength, the distribution pattern can be utilized as a load spectrum. Illustration of these applications will be given in a later section.

DERIVATION OF DISTRIBUTIONS OF SHIP MOTIONS AND LONGITUDINAL BENDING MOMENTS OF THE HULL

It will be assumed without further discussion that the short-term distribution of wave-induced ship motions and stresses may be represented by the one-parameter Rayleigh distribution and that the corresponding long-term distributions are approximated by the two-parameter log-normal distribution. Evidence to support these hypotheses is presented in Reference 2.

Typical distribution patterns of variation* in pitch angle are shown in Figures 3 and 4. In all, 129 similar sets were analyzed. Pertinent results are given in Tables 1 through 5 for variations of pitch angles, pitch acceleration, roll angle, heave acceleration, and the hull girder stress in the main deck amidships due to bending of the ship in a longitudinal plane normal to the deck.

It is interesting to note that all cumulative Rayleigh distributions (for example, that shown in Figure 4) become identical if $v^2 = x^2/E$ is plotted against the probability instead of plotting x directly. Utilizing this artifice it is necessary to know only the value of E corresponding to a particular sea condition, ship speed, and heading in order to obtain the probability of exceeding any value of x from a single graph (Figure 4) which is equally applicable to wave heights, ship motions, and hull stresses. The values of E for various ship operations are given in Tables 1 through 5. Table 4 of Reference 2 gives factors which, together with the value E, permit making statistical predictions as discussed later.

We now proceed to utilize the short-term distributions, each of which is characterized by a value of E, as building blocks in order to construct the long-term frequency distribution patterns of the ship responses to the sea applicable to *wartime service in the North Atlantic Ocean.* (It should be noted that the distribution patterns for other "missions" can be readily computed from the data given in this report.) Each of these short-term distributions will be weighted in accordance with the relative fraction of time spent at given sea state (f_2), at the given heading to the sea (f_3), and at the given ship speed (f_1). For example, if tests have indicated

* Throughout this paper, a variation is taken to mean the peak-to-peak variation of the variable.

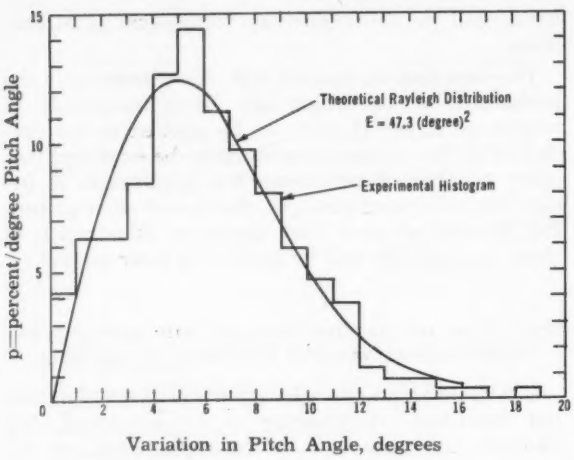


Figure 3. Distribution of Variation in Pitch Angle (Sample 1) for USCGC *Unimak*.

Conditions at time of test were: State 5 sea, head seas, ship speed 7½ knots, significant wave height 21 ft. The ship pitched at a rate of about 330 variations per hour. The test sample consisted of 236 variations in pitch angle. This sample is representative of 129 samples that were analyzed. Each such sample has been developed from analysis of oscillograms covering continuous periods of one-half hour or thereabout.

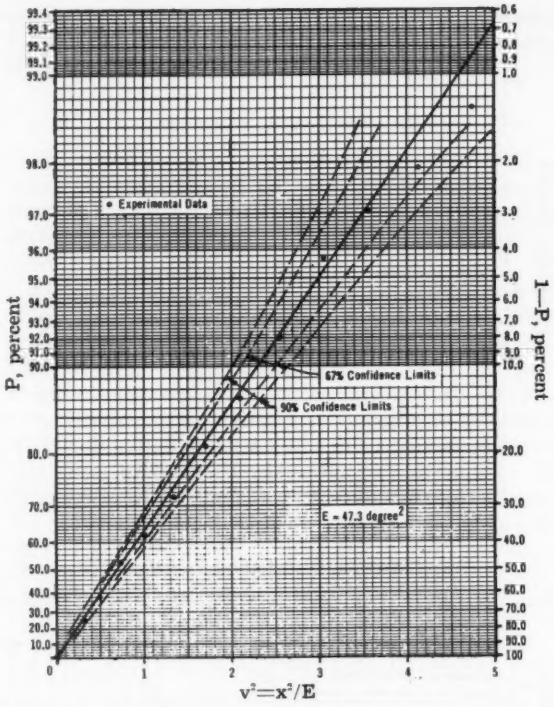


Figure 4. Cumulative Distribution of Variation in Pitch Angle (Sample 1) for USCGC *Unimak*.

Conditions at time of test were: State 5 sea, head seas, ship speed 7½ knots.

See Figure 3 for corresponding histogram. The plotted confidence limits do not include an allowance for errors of measurement.

that the ship will experience $N=670$ stretss variations per hour in a State 2 sea when heading directly into the waves at a speed of 10 knots, then one may expect that $n=f_1f_2f_3N=(0.33)\,(0.34)\,(0.125)\,670 = 9.40$ variations of stress per hour, out of the average number of variations per hour, can be attributed to this set of environmental conditions over an average year's operation in the assigned mission.

These calculations are carried out in Table 6 for variations in hull stress. See Reference 3 for similar calculations for pitch angle, roll angle, and pitch acceleration. Each horizontal line in this table gives the data corresponding to a given set of environmental conditions. The probabilities $(1-P)$ of exceeding given values of stress are computed and tabulated in columns 10 through 18. The total number of variations per hour which, over the average year, will exceed each given level are obtained by summing the product of column 9 with columns 10 through 18 over all environmental conditions. The last line in the table gives the probability of exceeding any one of the given magnitudes, for the long-term distribution. The latter values are plotted on the cumulative probability distribution charts in Figures 5 through 8.

The straight lines shown on these charts have *not* been drawn by eye through the plotted points but have been computed directly from the percentages represented by the plotted points under the assumption that the long-term distribution is of the log-normal type. A rather good fit of the computed line to the plotted points which is obtained indicates that this assumption is reasonable. One would expect that the points corresponding to the more extreme values would lie above the theoretical line because by far the greatest contribution to the computed probability for these extreme values derives from the more severe sea conditions. It is apparent that if data had been available for more severe seas than State 5, the probabilities of exceeding the higher values would have been increased whereas the plotted points representing probabilities of exceeding low or medium large values would not have been affected to any noticeable extent.

The value of E corresponding to any short-term distribution may readily be used to predict the most probable maximum value of the motion or stress expected in any given number of oscillations. Longuet-Higgins [8] has shown that the largest probable value out of N measurements is $\sqrt{E}$ times a constant if the population is of the Rayleigh type, where the constant is a function of N only. For large values of N, the constant is nearly equal to $\sqrt{\log_e N}$. Table 4 of Reference 2 gives the value of the constant by which $\sqrt{E}$ must be multipled. A

comparison of predicted and measured maximum values, utilizing this method, is given in Tables 1 through 5. There appears to be a satisfactory agreement.

The wave-induced hull-girder stresses can be converted to the corresponding vertical bending moments amidships by making use of the midship section modulus which is applicable to the strain-gage location (23.8 ft. above baseline, 10 ft. above the location of the neutral axis). Tests have indicated[1, 6] that the deckhouse of the AVP vessel is fully effective in resisting bending, thus resulting in a section moment of inertia of 761 ft.^{4} which corresponds to a section modulus applicable to the strain-gage location of 11,000 ft.-in^2. This value of the section modulus has been used to convert wave induced stresses to wave-induced bending moments.

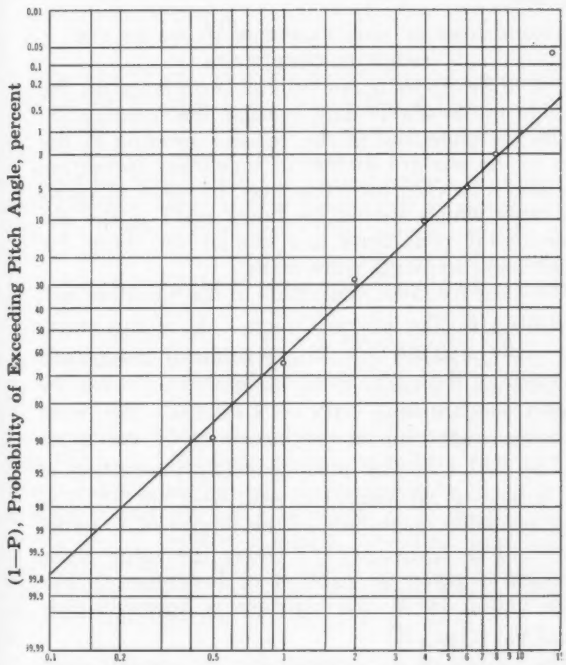


Figure 5. Long-Term Cumulative Distribution of Pitch Angle for Wartime Service, North Atlantic Ocean.

This log-normal distribution is computed on the basis of the data represented by the plotted points and it corresponds to the following parameters:

Mean value of $\log_{10}$ (variation in pitch angle)=0.1227

Standard deviation of $\log_{10}$ (variation in pitch angle)= 0.3898

It is probably valid up to 25 deg.

DESIGN AND OPERATIONAL CONDITIONS FOR WARTIME SERVICE

In the discussion of the statistical background, it was pointed out that the distribution patterns readily give the probability of exceeding any given mag-

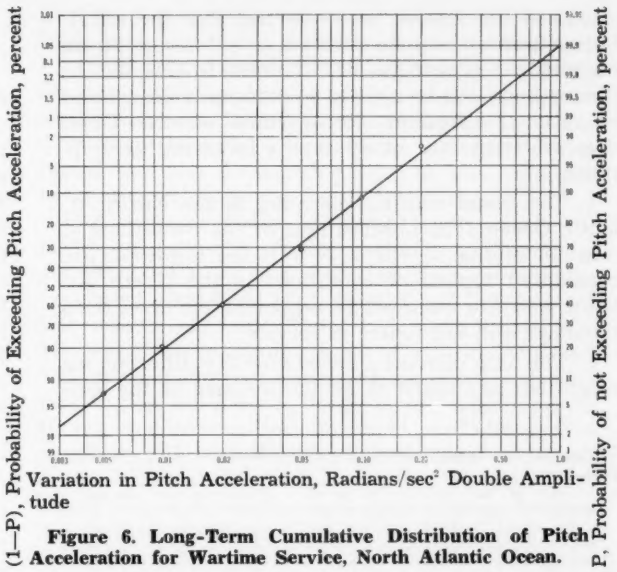

(1—P), Probability of Exceeding Pitch Acceleration, percent
P, Probability of not Exceeding Pitch Acceleration, percent
Variation in Pitch Acceleration, Radians/sec^2 Double Amplitude

Figure 6. Long-Term Cumulative Distribution of Pitch Acceleration for Wartime Service, North Atlantic Ocean.

This log-normal distribution is computed on the basis of the data represented by the plotted points and it corresponds to the following parameters:

Mean value of $\log_{10}$ (pitch acceleration in rad/sec^2)=2.4314

Standard deviation of $\log_{10}$ (pitch acceleration in rad/sec^2) =0.4895

It is probably valid up to 0.80 rad/sec^2.

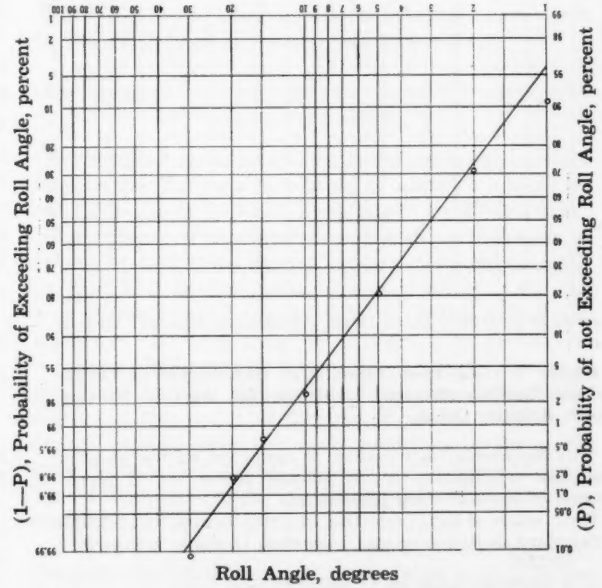


Figure 7. Long-Term Cumulative Distribution of Roll Angle for Wartime Service, North Atlantic Ocean.

This log-normal distribution is computed on the basis of the data represented by the plotted points and it corresponds to the following parameters:

Mean value of $\log_{10}$ (variation in roll angle in degrees)= 0.4868

Standard deviation of $\log_{10}$ (variation in roll angle in degrees)=0.2714

It is probably valid up to 56 deg.

nitude of the motion or stress and that the distribution pattern can be utilized as a load spectrum for endurance strength calculations. In this section the methods will be applied to determine design and operational conditions for wartime service. These determinations are based on the following assumptions:

1. The vessel will be operating in the North Atlantic Ocean. The observations of sea conditions at weather station C (52°N 37°W), see Figure 2, are considered typical of conditions in the North Atlantic and are assumed to represent the conditions the ships will encounter in service.

2. The ship operating speeds for the different sea conditions are taken to be the average of the esti-

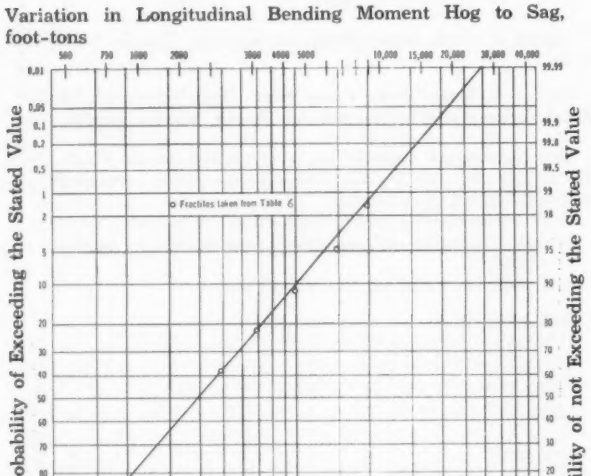


Figure 8. Long-Term Cumulative Distribution of Longitudinal Bending Moment, Amidships, for Wartime Service, North Atlantic Ocean.

This log-normal distribution is computed on the basis of the data represented by the plotted points and it corresponds to the following parameters:
Mean value of $\log_{10}$ (variation in stress in kips/in.^{2})=1.6016
Standard deviation of $\log_{10}$ (variation in stress in kips/in.^{2}) =0.3229
It is probably valid up to 40,000 ft-tons.

mates obtained from ship's officers of a number of ships.

3. The headings of the ship relative to the predominant wave direction are assumed equally likely, excepting only that seas coming approximately off the beam are considered unlikely for combinations of high speeds and rough seas.

"LONG-TERM" DISTRIBUTIONS OF SHIP MOTION, HULL BENDING MOMENT, AND WAVE HEIGHT

The charts of Figures 5 through 8 give the probability of exceeding and of not exceeding any given value of stress or motion if all the motions or stresses are considered to which the vessel is subjected over a period of a number of years. For example, only 3 per cent of all variations in roll angle would, on the average, exceed a value of 10 degrees peak-to-peak; see Figure 7. Figure 2 gives similar data for significant* wave heights to be expected in the North Atlantic Ocean. These distributions may be considered valid up to maximum variations of 25 degrees in pitch, 56 degrees in roll, 0.8 rad./sec.^{2} in pitch acceleration, and 40,000 ft.-tons in bending moment.

PREDICTIONS OF SHIP RESPONSE TO WAVES FOR SPECIFIED CONDITIONS

It is difficult to make reliable estimates on the basis of the available data because the specification of the sea state and of the relative heading of the ship to the sea are somewhat indefinite. Nevertheless the following formulas may be used to make estimates which should be fairly good if the environmental conditions are similar to those for which measurements were made.

The most frequent magnitude of the variation will be $0.707\sqrt{E}$. The average magnitude of the variation will be $0.886\sqrt{E}$. The significant magnitude (average of the upper third of the waves having the largest magnitudes) will be $1.416\sqrt{E}$. The most probable magnitude of the largest of 50 variations will be $2.01\sqrt{E}$. The most probable magnitude of the largest of 500 variations will be $2.509\sqrt{E}$. The most probable magnitude of the largest of N variations will be (constant) $\sqrt{E}$ where the value of the constant is given in Table 4 of Reference 2. For large values of N, the constant is approximately equal to $\sqrt{\log_e N}$.

The values of E corresponding to various combinations of sea condition, ship speed, and heading are given in Tables 1 through 5.

PREDICTION OF EXTREME VALUES

It may be desired to estimate the largest value of ship motions or hull girder stress that a ship structure is expected to experience in a given number of variations or over a given period of time.** The cumulative probability functions plotted in Figure 4 (Rayleigh distribution) and Figures 5 through 8 (log-normal distributions) may be used to find the probability of exceeding (1-P) or of not exceeding (P) a given value of the variable.

In the preceding section a convenient formula is given for estimating the most probable magnitude of the largest variation out of N variations; this formula is applicable only when the conditions of

* The significant wave height is used to denote a sea state. It is estimated as indicated in the first footnote on page 378.
** The average number of variations expected per hour at sea operation are given in Tables 1 through 5, column 5.

ship speed, heading, speed, and sea are steady. The following is a more general approach which may be applied to any distribution provided that its cumulative distribution function $P(x)$ is known.

The following formula, developed in Reference 2, gives the fraction f of all samples of size N [belonging to a given distribution specified by $P(x)$] which will have at least one value of $x > x_{m_1}$. The formula is valid only if $P(x_{m_1})$ has a value close to unity, that is, the formula is designed to estimate the values which occur rarely. Th eformula is

$$f = 1 - e^{-N[1-P(x_{m_1})]}$$

or alternatively

$$1 - P(x_{m_1}) = \frac{-\log_e(1-f)}{N}$$

where $[1 - P(x_{m_1})]$ is the probability of exceeding x_{m_1} in the fraction f of all samples of size N each. Knowing $[1 - P(x_{m_1})]$, it is easy to compute the corresponding value of the variable x_{m_1}.

In order to estimate the largest values of motion and bending moments for design purposes we will use this formula to estimate the value x_{m_1} which, on the average, is exceeded by the fraction f of all similar ships during their service life. Thus f represents the risk of exceeding x_{m_1}.

It will be assumed that the worst combination of operating conditions is the most severe of those listed in Tables 1 through 5, viz., a State 5 sea characterized by a significant wave height estimated to be 21 ft. The values of E_m specifying the corresponding Rayleigh distributions are listed in Table 7. Assume that the ship will be subjected to these operating conditions for a duration of 12 hours, experiencing V variations in this period of time, and that this situation will be repeated n times during the service life of the ship; therefore $N = nV$. For the Rayleigh distribution we have

$$[1 - P(x_{m_1})] = e^{-x_{m_1}^2/E_m}$$

Substitution in the expression for f gives:

$$1 - f = \exp[-e^{-y}] \text{ where } y = \frac{x_{m_1}^2}{E_m} - \log_e N$$

Table 1 of Reference 9 tabulates the values of exp $[-e^{-y}]$ as a function of y. Thus, for a specified risk f of exceeding x_{m_1} one may look up the corresponding value of y and then solve for the desired value of x_{m_1} from the relation

$$x_{m_1}^2 = E_m [y + \log_e N]$$

As an example consider the maximum value for the variation in roll angle.

From Table 7: $E_m = 176$ (deg.)^2, $V = 4600$

If we take $f = 0.001$ and $n = 10$, then Reference 9 gives $y = 7.0$. Therefore:

$$x_{m_1} = [176\ (7.0 + 8.43)]^{1/2} = 52 \text{ deg (port to starboard)}$$

Maximum estimated values for the other variables have been computed similarly, taking $f = 0.001$ and $n = 10$. They are listed in Table 7 together with the largest values measured at any time during the rough water sea trials reported in References 1 and 6. We will take the larger of the statistically estimated and the measured values as the suggested maximum value to use for design purposes.

For the same design problems, it is necessary to make an estimate of the extreme conditions of simultaneous pitch and roll. It is unlikely that the maximum conditions of pitch and roll listed in the last column of Table 7 will occur simultaneously. The most severe combinations of simultaneous roll and pitch angle, measured during the sea trials [1] was 40 deg. double amplitude in roll together with 10 deg. double amplitude in pitch, see Table 7. It is suggested that this combination be used as an extreme condition for design purposes. Predicted extreme values should be used with caution because the method eventually breaks down by predicting too extreme a value. This occurs because, in practical application the theoretical distribution cannot be relied upon at the extreme ranges of the function. For example, the prediction of 32 deg. for the extreme value of pitch angle variation is probably unrealistic as nonlinear behavior of the pitch motion will probably set a lower limit than this.

The extreme values listed in the last column of Table 7 may be used to set an interim upper limit to the validity of predicted extreme values.

DESIGN LOADS FOR BOTTOM STRUCTURE TO WITHSTAND SLAMMING LOADS

A detailed analysis of the loads, stresses, and deflection for the bottom plating of an AVP vessel incident to slamming has been made by Greenspon [5] On the basis of the latter study and photographs of an AVP during slamming,[1] it is estimated that the part of the bottom structure extending from the keel to the turn of the bilge in the forward quarter length of the ship may be subjected to occasional localized slamming or pounding pressures. In some locations within this area, the pressure attains values of the order of 300 psi. For the purpose of design calculations, the time variation of the pressure is such that it may be assumed to act statically. Thet following design loads are given as in interim recommendation:

a. For the design of bottom panels: Assume that the pressure attains a maximum value which varies linearly from 300 psi at the bow to 30 psi at a distance of 0.25 L from the bow:

$$p_0 = 300 - 1080\frac{x}{L}, \quad \frac{x}{L} < 0.25$$

where p_0 is the design pressure for plates,
x is the location of the plate measured from bow in feet ,and
L is the length of the ship in feet.

b. For the design of transverse framing: Assume that a static uniform pressure of intensity $p_0/2$ acts on the bottom over a strip one frame space in width.

c. For the design of longitudinal stiffeners or framing: Assume that a static pressure acts on the bottom plating supported by the longitudinal equal to $p_0/2$.

If the bottom plating is to be designed to keep the stresses within the elastic limit, then the simple formulas and tables of Reference 10 may be conveniently used for the calculation of the maximum stresses and deflections in plates loaded transversely. The ultimate load for transversely loaded panels may be estimated by utilizing the simple method given by Greenspon.[1]

SAMPLE OSCILLOGRAMS

The response to slamming is indicated in the oscillograms of pitch acceleration and hull girder stress amidships in Figure 9. The maximum peak-to-peak variations in stress measured in the keel fifteen inches aft of Frame 23 were 3500 psi, associated with presently undefined higher modes of hull structure motion which were excited subsequent to slamming at the bow.

DISCUSSION

The data of Tables 1 through 5 furnish a basis for working up distributions of motion and hull bending moment for assigned service missions of ships of this type. In the present instance, these basic data have been utilized to predict wartime service conditions for operation in the North Atlantic; the procedure for making this prediction is carried out in Table 6 as a guide for similar analysis of other missions. It should be pointed out that the prediction for "Wartime North Atlantic" service is greatly influenced by the estimated operational speeds provided by the U.S. Coast Guard.

It should be emphasized that the sea state is a variable which is most difficult to define. In the present case, variability of the sea conditions, for a given sea state, has been minimized by conducting the tests for a given sea state during one continuous time interval with the exception of the data for the State 3 Sea, for which two different time periods were required. The estimates of the sea state, as defined by the scale given in Reference 4, were made by the Weather Bureau observers who were assigned to the USCGC *Unimak* during the trials.

Inasmuch as the evaluation of sea states by experienced weather observers during the sea tests depended upon visual observations, one may expect reasonable agreement in the severity of the sea as defined by the sea state. In any case, no better basis for making the synthesis given in Table 6 was available to the authors.

The general method of synthesis utilized in this report could also be applied to data obtained from model tests of ships in waves rather than from full-scale test data. This would be a more flexible arrangement in that a wider variety of conditions could economically be covered by model tests than would be feasible with full-scale tests. Furthermore, the problem of measurement would be less difficult.

The order of magnitude of the pressures measured on the USCGC *Unimak* were also experienced by a sister ship, the USCGC *Casco* (formely AVP 12), during earlier sea tests in 1951. The indications are that the measured pressures are typical of the loading to be expected in heavy seas for this type of ship.

ACKNOWLEDGMENTS

The cooperation of the Commandant, U.S. Coast Guard and the Commanders, Air Force, U.S. Pacific and U.S. Atlantic Fleets made it possible to obtain realistic operational estimates of the speeds and headings under which ships of the AVP class are expected to operate in service. Mr. B. M. Wigle of the Vibrations Division, Taylor Model Basin, assited in the calculations.

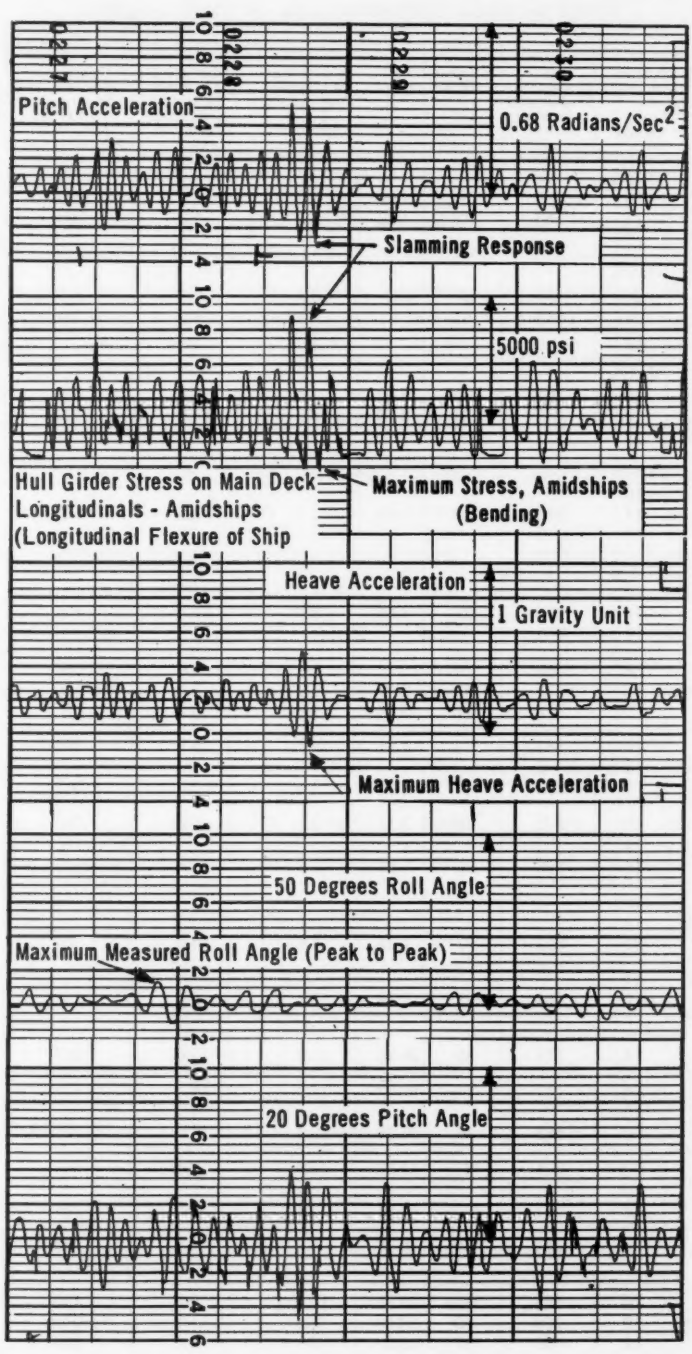

Pitch Acceleration
0.68 Radians/Sec²
Slamming Response
5000 psi
Hull Girder Stress on Main Deck Longitudinals - Amidships (Longitudinal Flexure of Ship
Maximum Stress, Amidships (Bending)
Heave Acceleration
1 Gravity Unit
Maximum Heave Acceleration
50 Degrees Roll Angle
Maximum Measured Roll Angle (Peak to Peak)
20 Degrees Pitch Angle

Figure 9. Oscillogram at Time of Maximum Measured Stress and Heave Acceleration 1 Feb. 1955.

Maximum Stress	5900 psi	Ships Speed	14.2 knots
Maximum Heave Acceleration	0.55 g	Relative Heading of Seas	015 deg.
Significant Wave Height	20 ft.	Direction from Which Waves Come	090 deg.
Wind Velocity	32 knots	Heading of Ship	075 deg.
Wind Direction	079 deg.		

TABLE 1—*Basic Statistical Data on Pitch Angles*

Sea State (Est'd)	Significant Wave Height ft.	Heading of Waves Relative to Ship	Ship Speed knots*	*N* Number of Variations per Hour	Minutes Sampled	*E* deg.²	Predicted Maximum Value for 1-hr. Operation	Maximum Measured Peak-to-Peak Variation deg.	Number of Variations in Sample from Which Maximum Was Obtained	Predicted Maximum Peak-to-Peak Variation	Ratio Predicted Maximum to Measured Maximum
2	6	Head Seas	7-7½								
			10	480	30	4.00	5.0	4.8	240	4.68	0.98
			14	555	32	4.48	5.3	5.3	296	5.04	0.95
			17								
2	6	Quarter Head Seas	7-7½								
			10	404	37	1.97	3.4	3.1	249	3.3	1.06
			14	514	32	1.86	3.5	3.3	274	3.24	0.98
			17								
2	6	Beam Seas	7-7½								
			10	561	29½	5.40	5.9	6.2	276	5.5	0.89
			14	643	32	3.75	4.9	4.8	343	4.69	0.98
			17								
2	6	Quarter Following Seas	7-7½								
			10								
			14	583	29	1.68	3.3	3.5	282	3.08	0.88
			17								
2	6	Following Seas	7-7½								
			10								
			14	578	32	1.86	3.5	3.8	308	3.27	0.86
			17								
3	7-9	Head Seas	10	583	26	10.35	8.1	7.6	253	7.56	0.99
			14	603	32	9.62	7.8	7.3	321	7.49	1.03
			17	616	27	9.59	7.9	6.7	277	7.34	1.09
3	7-9	Quarter Head Seas	10	587	27½	1.79	3.4	3.1	269	3.16	1.02
			14								
			17	592	37	2.02	3.6	3.9	365	3.45	0.89
3	7-9	Beam Seas	10	542	29	3.52	4.7	4.3	262	4.44	1.03
			14								
			17	298	26	1.08	2.5	3.3	129	2.29	0.70
3	7-9	Quarter Following Seas	10								
			14								
			17	523	31	1.85	3.4	3.9	270	3.16	0.85
3	7-9	Following Seas	14								
			17	504	23	1.28	2.8	2.4	193	2.59	1.08
4	16	Head Seas	14	578	30	26.12	12.9	11.6	289	12.1	1.00
5	21	Head Seas	7½	517	27½	47.32	17.2	18.0	236	16.1	0.90
			10								
			14	583	24	56.00	18.9	15.5	235	17.4	1.16
5	21	Quarter Head Seas	7½	541	38½	35.13	14.9	13.9	347	14.3	1.03
			10								
			14	577	28½	47.14	17.3	16.5	274	16.3	0.99
5	21	Beam Seas	7½	364	30	16.78	9.9	8.8	182	9.34	1.06
5	21	Quarter Following Seas	7½	368	28½	20.24	10.9	9.8	175	10.2	1.04
			10	296	29	28.00	12.6	11.5	143	11.8	1.02
			14	302	27	17.2	9.9	8.8	136	9.16	1.04
5	21	Following Seas	7½	291	33	31.12	13.3	11.8	160	12.58	1.06
			10								
			14	273	31	20.54	10.8	10.8	140	10.1	0.94

* The speed is the nominal speed read from a calibration curve of propeller rpm versus knots; 7½, 10, 14, and 17 knots correspond to 94, 127, 185, and 230 rpm, respectively.

TABLE 2—*Basic Statistical Data on Pitch Accelerations*

Sea State (Est'd)	Significant Wave Height ft.	Heading of Waves Relative to Ship	Ship Speed knots*	*N* Number of Variations per Hour	Minutes Sampled	$E\left(\frac{\text{rad.}}{\text{sec}^2}\right)^2$	Predicted Maximum Value for 1-hr. Operation	Maximum Measured Peak-to-Peak Variation rad/sec.^{2}	Number of Variations in Sample from Which Maximum Was Obtained	Predicted Maximum Peak-to-Peak Variation	Ratio Predicted Maximum to Measured Maximum
		Head	10	704	30	0.0015	0.099	0.093	352	0.093	1.00
2	6	Seas	14	786	32	0.0024	0.126	0.123	419	0.121	0.99
		Quarter	10	761	32	0.0015	0.100	0.093	406	0.095	1.02
2	6	Head Seas	14	872	32	0.0014	0.097	0.097	465	0.093	0.960
		Beam	10	708	29½	0.0033	0.147	0.144	348	0.139	0.97
2	6	Seas	14	740	32	0.0033	0.148	0.140	395	0.140	1.00
2	6	Quarter Following Seas	14	890	29	0.0016	0.104	0.11	430	0.099	0.90
2	6	Following Seas	14	635	32	0.00028	0.043	0.047	339	0.0404	0.86
			10	694	26	0.0109	0.267	0.28	301	0.250	0.90
3	7-9	Head Seas	14	765	32	0.0161	0.327	0.322	396	0.310	0.97
			17	862	27	0.0171	0.341	0.357	388	0.319	0.90
		Quarter	10	779	27½	0.0022	0.121	0.13	357	0.114	0.88
3	7-9	Head Seas	17	900	37	0.0028	0.138	0.15	555	0.133	0.89
		Beam	10	763	29	0.0030	0.142	0.161	369	0.133	0.83
3	7-9	Seas	17	760	26	0.0029	0.139	0.174	329	0.130	0.75
3	7-9	Quarter Following Seas	17	696	31	0.001	0.081	0.075	338	0.076	1.00
3	7-9	Following Seas	17	629	23	0.00072	0.068	0.064	241	0.063	0.98
4	16	Head Seas	14	780	30	0.0222	0.384	0.39	390	0.35	0.90
			7½	624	27½	0.0173	0.334	0.425	284	0.341	0.80
5	21	Head Seas	14	692	24	0.0308	0.450	0.47	278	0.42	0.89
		Quarter	7½	615	38½	0.0140	0.300	0.314	395	0.29	0.92
5	21	Head Seas	14	711	28½	0.0327	0.465	0.47	338	0.437	0.93
5	21	Beam Seas	7½	458	30	0.0039	0.155	0.161	229	0.145	0.90
		Quarter	7½	702	28½	0.0032	0.145	0.146	333	0.136	0.93
5	21	Following	10	660	29	0.0023	0.123	0.129	319	0.115	0.89
		Seas	14	714	27	0.0023	0.123	0.131	322	0.115	0.88
5	21	Following	7½	524	33	0.0013	0.090	0.068	288	0.085	1.25
		Seas	14	470	31	0.0004	0.050	0.058	243	0.047	0.81

* The speed is the nominal speed read from a calibration curve of propeller rpm versus knots; 7½, 10, 14, and 17 knots correspond to 94, 127, 185, and 230 rpm, respectively.

TABLE 3—*Basic Statistical Data on Roll Angles*

Sea State (Est'd)	Significant Wave Height ft.	Heading of Waves Relative to Ship	Ship Speed knots*	*N* Number of Variations per Hour	Minutes Sampled	*E* deg.²	Predicted Maximum Value for 1-hr. Operation	Maximum Measured Peak-to-Peak Variation deg.	Number of Variations in Sample from Which Maximum Was Obtained	Predicted Maximum Peak-to-Peak Variation	Ratio Predicted Maximum to Measured Maximum
		Head	10	380	30	19.0	10.7	9.4	190	9.96	1.06
2	6	Seas	14	362	32	18.3	10.4	10.5	193	9.80	0.93
		Quarter Head	10	355	37	31.0	13.5	13.1	219	12.9	0.99
2	6	Seas	14	340	32	26.0	12.4	12.2	191	11.7	0.96
		Beam	10	388	29½	17.8	10.3	10.6	191	9.66	0.91
2	6	Seas	14	368	32	19.0	10.6	10.3	196	10.0	0.97
2	6	Quarter Following Seas	14	335	29	24.0	11.9	10.9	172	11.1	1.02
2	6	Following Seas	14	326	32	21.0	11.0	10.3	174	10.4	1.01
		Head	10	436	26	20.6	11.2	10.6	189	10.4	0.98
3	7-9		14	400	32	17.5	10.3	9.7	213	9.66	1.00
		Seas	17	448	27	9.8	7.75	7.2	202	7.23	1.00
		Quarter Head	10	209	27½	6.5	6.0	6.0	96	5.45	0.91
3	7-9	Seas	17	339	37	7.5	6.6	6.7	209	6.34	0.95
		Beam	10	260	30	6.9	6.2	8.7	130	5.8	0.67
3	7-9	Seas	17	482	26	13.0	9.0	9.2	209	8.35	0.91
3	7-9	Quarter Following Seas	17	285	31	19.3	10.5	10.9	147	9.8	0.90
3	7-9	Following Seas	17	355	23	15.8	9.6	9.1	136	8.81	0.97
4	16	Head Seas	14	348	30	7.8	6.8	6.9	174	6.35	0.92
		Head	7½	369	27½	119.	26.6	24.4	168	24.7	1.01
5	21	Seas	14	398	24	84.8	22.5	20.6	160	20.7	1.01
		Quarter Head	7½	388	38½	75.3	21.2	18.1	249	21.0	1.16
5	21	Seas	14	408	28½	64.3	19.8	23.0	194	18.4	0.80
5	21	Beam Seas	7½	364	30	171.	31.8	25.0	182	29.9	0.84
		Quarter	7½	384	28½	176.	32.4	38.8	182	30.2	0.78
5	21	Following	10	372	29	148.	29.6	24.0	180	27.8	1.15
		Seas	14	360	27	114.	25.9	21.9	162	24.1	1.10
		Following	7½	356	33	150.	29.6	24.4	196	28.1	1.15
5	21	Seas	14	340	31	60.	18.8	18.5	176	17.6	0.95

* The speed is the nominal speed read from a calibration curve of propeller rpm versus knots; 7½, 10, 14, and 17 knots correspond to 94, 127, 185, and 230 rpm, respectively.

TABLE 4—*Basic Statistical Data on Stresses*

Sea State (Est'd)	Significant Wave Height ft.	Heading of Waves Relative to Ship	Ship Speed knots*	*N* Number of Variations per Hour	Minutes Sampled	$\left(\frac{E}{\text{kips/in.}^2}\right)^2$	Predicted Maximum Value for 1-hr. Operation kips	Maximum Measured Peak-to-Peak Variation kips/in.^{2}	Number of Variations in Sample from Which Maximum Was Obtained	Predicted Maximum Peak-to-Peak Variation	Ratio Predicted Maximum to Measured Maximum
2	6	Head Seas	10	670	30	0.22	1.20	1.2	335	1.13	0.94
			14	784	32	0.31	1.44	1.5	418	1.37	0.91
2	6	Quarter Head Seas	10	256	37	0.21	1.08	1.2	158	1.03	0.86
			14	1004	32	0.24	1.30	1.3	536	1.22	0.94
2	6	Beam Seas	10	522	29½	0.37	1.52	1.6	257	1.43	0.89
			14	666	32	0.29	1.38	1.7	355	1.31	0.77
2	6	Quarter Following Seas	14	640	27	0.22	1.19	1.5	288	1.11	0.74
2	6	Following Seas	14	392	32	0.36	1.46	1.4	209	1.39	0.99
3	7–9	Head Seas	10	621	26	1.12	2.70	2.6	269	2.5	0.96
			14	778	32	1.28	2.92	2.8	414	2.84	1.01
			17	991	27	1.21	2.88	2.9	446	2.72	0.94
3	7–9	Quarter Head Seas	10	864	27½	0.11	0.86	0.9	396	0.81	0.90
			17	711	37	0.16	1.02	1.0	438	0.985	0.99
3	7–9	Beam Seas	10	503	29	0.19	1.09	0.94	243	1.02	1.08
			17	685	26	0.18	1.09	0.97	297	1.01	1.04
3	7–9	Quarter Following Seas	17	459	31	0.15	0.96	1.0	237	0.905	0.91
3	7–9	Following Seas	17	360	23	0.09	0.72	0.59	138	0.666	1.13
4	16	Head Seas	14	768	30	1.17	2.8	3.1	384	2.64	0.85
5	21	Head Seas	7½	581	27½	2.81	4.2	3.9	265	3.96	1.02
			14	790	24	3.50	4.8	4.5	318	4.48	1.00
5	21	Quarter Head Seas	7½	598	38½	2.34	3.9	3.9	384	3.73	0.96
			14	694	28½	3.00	4.4	4.0	330	4.16	0.96
5	21	Beam Seas	7½	532	30	1.3	2.8	2.7	266	2.69	1.00
5	21	Quarter Following Seas	7½	449	28	0.86	2.3	2.0	212	2.14	0.94
			10	328	29	1.12	2.5	2.3	159	2.38	1.03
			14	344	27	1.01	2.4	2.3	155	2.25	0.98
5	21	Following Seas	7½	325	33	1.90	3.3	3.6	179	3.14	0.87
			14	300	31	1.57	3.0	3.1	155	2.82	0.91

* The speed is the nominal speed read from a calibration curve of propeller rpm versus knots; 7½, 10, 14, and 17 knots correspond to 94, 127, 185, and 230 rpm, respectively.

TABLE 5—*Basic Statistical Data on Heave Accelerations*

Sea State (Est'd)	Significant Wave Height ft.	Heading of Waves Relative to Ship	Ship Speed knots*	*N* Number of Variations per Hour	Minutes Sampled	*E* g's²	Predicted Maximum Value for 1-hr. Operation g's	Maximum Measured Peak-to-Peak Variation g's	Number of Variations in Sample from Which Maximum Was Obtained	Predicted Maximum Peak-to-Peak Variation	Ratio Predicted Maximum to Measured Maximum
3	7-9	Head	10	448	26	0.0107	0.252	0.26	239	0.251	0.97
		Seas	14	590	32	0.0186	0.35	0.32	315	0.321	1.01
			17	582	27	0.0221	0.37	0.37	262	0.35	0.95
5	21	Head	7½	433	27	0.0221	0.37	0.51	197	0.341	0.67
		Seas	10	518	27	0.0272	0.41	0.41	233	0.385	0.94
			14	514	24½	0.0382	0.49	0.49	210	0.451	0.92
5	21	Quarter	7½	466	38½	0.0182	0.33	0.29	299	0.322	1.11
		Head	10	583	28	0.0245	0.40	0.44	272	0.371	0.84
		Seas	14	565	27½	0.0498	0.57	0.62	259	0.525	0.85
5	21	Beam Seas	7½	433	28	0.0143	0.29	0.26	202	0.276	1.06
5	21	Quarter Following	7½	418	32	0.0147	0.30	0.35	223	0.282	0.81
			10	552	30	0.0131	0.29	0.34	226	0.267	0.78
		Seas	14	504	27	0.0135	0.29	0.32	227	0.270	0.84

* The speed is the nominal speed read from a calibration curve of propeller rpm versus knots; 7½, 10, 14, and 17 knots correspond to 94, 127, 185, and 230 rpm, respectively.

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TABLE 6

Derivation of Predicted Distribution Pattern for Variation in Hull Stress Due to Longitudinal Bending, at the Main Deck, Amidships for Wartime Duty in North Atlantic Ocean

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18
Sea State	Significant Wave Height* ft.	Relative Heading of Waves	Ship Speed** knots	Weighting Factors†				Average Number of Variations per Hour Contributed by Each Operating Condition	Probability of Exceeding Given Magnitude of Variation††(1 - P) Kips per Square Inch								E (kips/in.²)² Mean Square Variations
				f_1	f_2	f_3	N	$f_1f_2f_3N=n$	0.2	0.5	1.0	1.5	2.0	4.0	8.0	0.7	
	<4				0.20		646	129.24	0.69541	0.1031	0.00011						0.11
2	6	H	10^7	0.33	0.34	0.125	670	9.40	0.835	0.321	0.0106	0.0000358	0.0123×10^{-6}	0.025×10^{-30}		0.108	0.22
		H	14^{17}	0.67			784	22.32	0.880	0.447	0.0398	0.000705	2.52×10^{-6}	0.038×10^{-21}		0.206	0.31
		QH	10^7	0.33		0.25	256	7.18	0.826	0.304	0.0094	0.0000250	0.00525×10^{-6}			0.097	0.21
		QH	14^{17}	0.67			1004	57.18	0.846	0.353	0.0156	0.0000855	0.057×10^{-6}			0.130	0.24
		B	10^7	0.33			522	14.64	0.898	0.509	0.0672	0.00228	20.7×10^{-6}	0.17×10^{-18}		0.266	0.37
		B	14^{17}	0.67			666	37.93	0.871	0.422	0.0320	0.000430	1.12×10^{-6}	1.0×10^{-24}		0.184	0.29
		QF	$14^{7,\ 10,\ 17}$	1.00			640	54.40	0.835	0.321	0.0106	0.0000358	0.0123×10^{-6}	0.025×10^{-30}		0.108	0.22
	↓	F	$14^{7,\ 10,\ 17}$	1.00	↓	0.125	392	16.66	0.896	0.495	0.0622	0.00192	15.0×10^{-6}	0.049×10^{-18}		0.256	0.36
3	7-9	H	10^7	0.30	0.29	0.125	621	6.75	0.96487	0.800	0.4095	0.1343	0.0280	0.624×10^{-6}		0.646	1.12
		H	14	0.38			778	10.72	0.96918	0.8227	0.4580	0.1725	0.0439	3.72×10^{-6}		0.672	1.28
		H	17	0.32			991	11.50	0.96749	0.8135	0.4378	0.1555	0.0368	1.82×10^{-6}		0.667	1.21
		QH	10^7	0.30		0.25	864	18.79	0.69541	0.1031	0.000113	1.30×10^{-9}	0.158×10^{-15}			0.0116	0.11
		QH	17^{14}	0.70			711	36.09	0.77881	0.2095	0.00192	0.78×10^{-6}	0.014×10^{-9}			0.0467	0.16
		B	10^7	0.30			503	10.94	0.81037	0.2681	0.00520	7.2×10^{-6}	0.72×10^{-9}			0.076	0.19
		B	17^{14}	0.70			685	34.76	0.80064	0.2495	0.00388	3.75×10^{-6}	0.238×10^{-9}			0.066	0.18
		QF	$17^{7,\ 10,\ 14}$	1.00			459	33.28	0.76610	0.1890	0.00128	0.308×10^{-6}	2.63×10^{-12}			0.038	0.15
	↓	F	$17^{7,\ 10,\ 14}$	1.00	↓	0.125	360	13.05	0.64123	0.0621	0.000015	0.0138×10^{-9}	0.048×10^{-18}			0.0043	0.09
4	16	H	$14^{7,\ 10,\ 17}$	1.00	0.11	1.00	768	84.48	0.96637	0.80775	0.4255	0.1460	0.0328	1.16×10^{-6}		0.658	1.17
5	21	H	7	0.36	0.06	0.125	581	1.57	0.98588	0.91474	0.7010	0.4490	0.2410	3.35×10^{-3}	0.125×10^{-9}	0.8400	2.81
		H	$14^{10,\ 17}$	0.64			790	3.80	0.98863	0.93096	0.7518	0.5260	0.3190	10.4×10^{-3}	11.6×10^{-9}	0.8692	3.50
		QH	7	0.36		0.25	598	-3.23	0.98301	0.90067	0.6525	0.3824	0.1810	1.08×10^{-3}	0.0013×10^{-9}	0.8112	2.34
		QH	$14^{10,\ 17}$	0.64		0.50	694	13.32	0.98675	0.91996	0.7167	0.4722	0.2640	4.83×10^{-3}	0.55×10^{-9}	0.8493	3.00
		B	7	0.36		0.25	532	2.87	0.96968	0.82508	0.4633	0.1773	0.0460	4.50×10^{-6}		0.6860	1.30
		QF	7	0.36			449	2.43	0.95453	0.74777	0.3125	0.0730	0.0096	8.2×10^{-9}		0.5660	0.86
		QF	10	0.28			328	1.38	0.96487	0.8000	0.4095	0.1340	0.0280	0.624×10^{-6}		0.646	1.12
		QF	14^{17}	0.36			344	1.86	0.96116	0.78064	0.3718	0.1078	0.0190	0.125×10^{-6}		0.6155	1.01
		F	7	0.36		0.125	325	0.88	0.97914	0.87657	0.5910	0.3060	0.1220	0.220×10^{-3}	1.52×10^{-15}	0.7725	1.90
	↓	F	$14^{10,\ 17}$	0.64	↓		300	1.44	0.97484	0.85280	0.5293	0.2385	0.0782	0.038×10^{-3}		0.7320	1.57
Number of Variations per Hour Exceeding Given Magnitude (Summing over all Conditions)								642.09	531.71	246.75	74.96	28.89	9.994	0.113		141.60	
Percentage of Variations Exceeding Given Level									82.8	38.43	11.67	5.00	1.56	0.0018		22.05	

* Column 2 gives the estimated significant wave height. This value was estimated by taking the mean value of the highest waves in the center of each group of well-formed waves.

** Column 4 gives the nominal speed taken from a calibration curve of shaft rpm versus knots. The exponents in this column represent ship speeds for which no data are available and for which the experimental data corresponding to the indicated speed are assumed to apply for purposes of the table.

†f_1 is the fraction of time the ship will operate at a given speed for a given sea state and includes all headings relative to the predominant wave direction.
f_2 is the fraction of time that a given sea state will be expected, on the average.
f_3 is the fraction of time the ship will make a given heading to the sea, for a given sea state and speed.
N is the number of variations expected per hour of operation under the condition considered. The value of N is taken from the tables of basic data.

†† The exponents in columns 10-18 denote the power of 10 by which the given number is to be multiplied.

TABLE 7—*Maximum Values of Ship Motion and Longitudinal Bending Moment for Use in Design Calculations*

All values given refer to the peak-to-peak variation.

Quantity	Condition for Which the Extreme Value is Predicted					V Number of Variations in 12 Hours for Steady Conditions	Estimated Most Probable Largest Variation in 12 Hours $\sqrt{E \log_e V}$	Maximum Variation for Operating Life of Ship $f=0.001$	Largest Measured Variation from Table 5, Ref. 2†	Maximum Variation for Design Purposes
	Sea State	Significant Wave Height	Direction of Seas	Shaft RPM	E_-					
Roll Angle	5	21	QF	94	176 (deg.)^2	4600	39°	52°	50°	56°
Pitch Angle	5	21	H	185	56 (deg.)^2	7000	22°	32°**	20°	about 25°
*Roll Angle	5	21	QF	94						40°
*Pitch Angle	5	21	QF	94						10°
Pitch Acceleration	5	21	QH	185	0.0327 (rad./sec.^{2})^2	8500	0.54 rad./sec.^{2}	0.78 rad/sec.^{2}	0.77 rad/sec.^{2}	0.8 rad/sec.^{2}
Heave Acceleration	5	21	QH	185	0.0498 (gravity unity)^2	6800	0.66 g	1.0 g	1.0 g	1.0 g
Longitudinal Bending Moment	5	21	H	185	85×10^6 (ft.-tons)^2	9500	28,000 ft.-tons	40,000 ft.-tons	33,000 ft.-tons	40,000 ft.-tons
Longitudinal Bending Stress	5	21	H	185	3.50 (kips/in.^{2})^2	9500	5.7 kips/in.^{2}	8.0 kips/in.^{2}	6.7 kips/in.^{2}	8.0 kips/in.^{2}

* This is believed to be a most severe condition of simultaneous roll and pitch. The data are taken from Figure 6b of Reference 1.
** This estimate (32°) is believed to be outside the range within which the statistical estimation is valid and therefore the value is discarded.
† These values are nearly the largest magnitudes obtained under any conditions experienced during the tests reported in Reference 1.
QH—Quarter Following; QF—Quarter Head; H—Head.

The Society has lost through death one of its members of longest standing.

CAPTAIN JOHN B. TURNER

joined the Society in 1903 and was one of those whom the Council named as Longevity Life Members when this action was taken a year ago in regard to all members who had retained their membership in good standing for 50 years or more.

Captain Turner was born on 18 February 1872. He joined the Revenue Cutter Service in 1895 and served on active duty for 35 years, 25 of which were on sea duty. He retired on 1 December 1930 as a Captain of Engineers of the Coast Guard which had been formed in 1915 by a merger of the Revenue Cutter Service and the Life Saving Service. All of his service was spent as an engineer.

During his 25 years of sea service Captain Turner served on many ships and through two wars. He served during the Spanish-American war on the U. S. Revenue Cutter *Grant* and for this received the Special Medal for Service. During World War I he was again at sea, this time on the U.S.C.G.C. *Blackhawk*. He received the Victory Medal for this service.

Captain Turner made his home after retirement at Rouses Point, New York, where he died on 30 April 1959. We have been unable to obtain any information as to his family survivors. His continuance of membership in the Society for 29 years after retirement from active duty seems to indicate that he maintain his interest in his service and in his engineering profession until his death.

DEATHS

Berner, George A., LCDR, USNR .....Naval
Homer, William Henry .....Associate
Johnson, Harvey F., RADM, USCG (Ret.) ...Honorary
Kaulback, Harold D., CAPT, USNR (Ret.) .....Naval
Sackman, John R., CDR, USNR .....Naval
Tummel, Harold E. Civil
Turner, John B., CAPT, USCG (Ret.) .....Life
Tyler, Gaines A., CDR, USCG .....Naval

HARVEY FLETCHER JOHNSON

Death has claimed a long time loyal member and past President of the Society.

Rear Admiral Harvey Fletcher Johnson, 76, U. S. Coast Guard, Retired, holder of the Legion of Merit for outstanding work as Engineer-in-Chief during World War II, and a veteran of World War I, died on 22 April 1959 of cancer of the throat at Bethesda Naval Hospital.

Admiral Johnson was elected and served as President of the Society in 1946 and was appointed to complete the term of Rear Admiral Roger Paine as President in 1948. He was the first officer of the Coast Guard to hold this office.

Rear Admiral Johnson was permanently assigned to Coast Guard Headquarters in Washington, D. C., in December 1932. He became Acting Engineer-in-Chief in October 1935, and served as Engineer-in-Chief from December 1935 until his retirement on August 1, 1946, after the close of World War II.

When awarded the Legion of Merit for his wartime tenure as Engineer-in-Chief, Admiral Johnson was cited as an astute and forceful administrator who organized and supervised a rigid program for the planning, building, and staffing of greatly expanded Coast Guard shore facilities both in combat areas and for the coastal defense of the United States. In addition he developed aids to navigation including Beacon stations and Loran projects, escort cutters and tender class vessels, planes, helicopters and numerous smaller craft used in Port Security work.

Also, as Chairman of the Merchant Marine Council and as President of a Board of Investigation to inquire into the design and construction of steel merchant vessels, he advised and guided the Merchant Marine and the Army Water Transportation Services in maintaining the safe and uninterrupted flow of enormous quantities of the nation's man-power and war materials to the battlefronts of the world.

During the war he was also Treasury Department representative of the Standards Council of the American Standards Association.

Admiral Johnson performed engineering duties at Coast Guard Headquarters in a lesser capacity on several other occasions, from September 1919 to September 1922; November 1922 to April 1924; and from March 1928 to February 1932.

He joined the Society in 1910 and when in Washington, particularly following retirement, he was most generous of his time and capability in serving the Society on various committees. He was a member of the Council in 1938.

He made his permanent home for the past 31 years at 115 Hesketh St., Chevy Chase, Maryland. He was born in Wheatley, Arkansas, on August 27, 1882. He received his early education in the public schools of New Orleans, Louisiana. After attending Louisiana State and Tulane Universities, he served for three years in the Merchant Marine of England.

He was graduated from Cornell University with a degree in Mechanical Engineering in 1906. He then worked for one year at the Newport News Shipbuilding and Drydock Company in Virginia.

In July 1907, he was appointed an engineering cadet in the Revenue Cutter Service, forerunner of the U.S. Coast Guard, and received a commission as Ensign on February 13, 1908. He became Rear Admiral on May 24, 1939.

His earlier Coast Guard assignments included engineering duties aboard the *Bear*, *Apache*, *Windom*, *Gresham*, and *Tybee*. During World War I he served in Europe as engineer officer aboard the USS *Aphrodite* and then the *Cleveland* (later named the *Mobile*), a German vessel transferred to the United States under terms of the Armistice to bring troops home from France.

Between ensuing earlier assignmnts at Coast Guard Headquarters, he served in various other stations. He was engineer officer of the destroyer *Jouett* from 1924 to 1926, then served at the Philadelphia Navy Yard in connection with reconditioning of other Navy destroyers transferred to the Coast Guard for use in the suppression of smuggling on the eastern seaboard. He subsequently served in the destroyer *Tucker*.

In 1932 he was Inspector of Machinery at Trenton, New Jersey, after which he was Chief Inspector at the Defoe Shipbuilding Company, Bay City, Michigan, during the construction of the Cutter *Escanaba*.

In addition to his membership in the American Society of Naval Engineers, Admiral Johnson was a member of the Society of Naval Architects and Marine Engineers, Cornell Engineers, District of Columbia Chapter of the Military Order of the World Wars, The Newcomen Society, Kappa Sigma Fraternity, Cornell Club of Washington, and the Columbia Country Club.

Admiral Johnson is survived by his wife, the former Eleanor Ann Walker of Baltimore, Maryland, a married son, Harvey Fletcher Johnson, Jr., of Sacramento, Calif.; a sister, Mrs. William Farrell of Baton Rouge, La.; two brothers, Vernon and Leandor Johnson of Hendersonville, N. C.

Burial with graveside services and full military honors was held at Arlington National Cemetery at 3:00 P.M., Friday, April 24.

BOOK REVIEWS & NOTICES

ANTI-COMMUNIST HANDBOOK
A GUIDE TO ANTI-COMMUNIST ACTION
by
Anthony Trawick Bouscaren
Henry Regnery Company, 1958
244 pages; $4.00
Reviewed by
Colonel James N. Peale, Jr.
Infantry, Chief of the Psychological Operations
Branch, Special Warfare Division, Plans Directorate,
Office of the Deputy Chief of Staff for Military
Operations, Department of the Army.

When Deputy Premier Anastas Mikoyan returned to Russia this January he reported that the Americans were favorably disposed towards "peaceful coexistence" and tired of the cold war and he concluded that acts must follow words. For the next several months Americans will hear the communists and their fellow travelers in America clamoring for a summit conference; trade and exchanges of visits; relaxation of vigilance by the Free World; and end of nuclear tests; abandonment of US bases overseas; dissolution of the Free World alliances; recognition of Red China; neutralization of Germany, Japan, Korea, and Vietnam; discontinuance of the message of hope to the people behind the Iron Curtain; and prudent retreat from the inevitable advance of communist influence and power. Recommended measures to prevent the progress of "Socialism" will be labled as dangerous and contrary to the spirit of peaceful coexistence. The agenda for world discussion will be announced by the enemies of the United States.

The citizen who is alarmed by this turn of affairs usually feels frustrated and helpless to stem the tide of opinion which flows contrary to the national interest. Anthony T. Bouscaren offers a way to overcome this frustration. He explains how to recognize communist propaganda, communists, and their front organizations and what a patriotic American can do in his own community to thwart the operations of this formidable conspiracy.

Bouscaren is well qualified on the subject. He is an associate professor of political science at Marquette University, now on leave of absence while serving on the faculty of the National War College. He was a bomber pilot in the Marine Corps during World War II and has visited many parts of Europe and Asia, written six books and numerous articles, lectured often, and conducted a citizens' seminar on communism.

Bouscaren explains peaceful coexistence as a strategem intended to lull the Free World into a state of passivity until the Soviet empire is again ready to resume the offensive,—as a means to "obtain at the conference table what they would otherwise have to obtain by force." He cites Stalin as saying of peaceful coexistence: "We are speaking of temporary agreements with the capitalist governments in the sphere of industry, trade, and perhaps diplomatic relations . . . [However] We are not living merely in a state, but in a system of states, and the existence of a Soviet Republic side by side with imperial states for a longer time is unthink-

able. One or the other must triumph . . . A series of frightful clashes between the Soviet Republic and the bourgeois is inevitable."

Bouscaren recommends that the United States "adopt liberation from communism as our long-range objective and convince the enslaved that we mean it." He feels that the U.S. government "should energetically sell anti-communism to its own people and to its allies." "Ours is the choice," he says, "to continue to lose 'World War III,' in which we have lost 8 million square miles of territory and 800 million people to Soviet expansion, not to mention 150,000 American casualties in Korea, through the defeatist coexistence policy; or to stir ourselves out of our lethargy immediately and take these effective steps, which alone can lead to ultimate victory and general peace."

In April 1955, he explains, Brigadier General David Sarnoff of RCA recommended a psychological offensive. Sarnoff wrote: "Our potential fifth columns are greater by millions than the enemy's. But they have yet to be given cohesion, direction, and the inner motive power of hope and the expectation of victory." The plan, which antedated the summit conference of 1955, was laid aside.

The reliance of the United States on the deterrent power of nuclear weapons and the diminution of the armies of Western Europe have created, he considers, a trend that is morally and politically to the advantage of the Soviet Union. Bouscaren recommends military establishment capable of fighting either nuclear wars or "brush fire wars," emphasis on research and development, and personnel reforms. He states regarding missile warfare, "The time in history is past when this nation can afford to let the enemy deliver the first blow."

The influence of communists in the United States is multiplied by front organizations, which are sustained by gullible intellectuals. These fellow travellers are often in a position to influence government, publishing houses, radio and television companies, industry, and schools and colleges where overt communist propaganda would be rejected. Mr. J. Edgar Hoover numbered these American front workers at 400,000,—or more than ten times the party membership. The Alien Registration Act of 1940 (Smith Act), the McCarran Internal Security Act of 1950, the Communist Control Act of 1954, and the Immunity Act of 1954 formed the legal basis of the Security Program, designed to curb communist influence. However, a number of supreme court decisions in 1957 greatly weakened the program and led the *Daily Worker* to say, "the curtain is closing on one of our worst periods."

Soviet espionage in the United States has been aided by *Amtorg*, a trade organization, by diplomats, and by reporters of *Tass* and *Pravda*, who are free to travel and use public communications facilities. Many officials have seemed oblivious to the evil motives of communists in government. In Bouscaren's opinion, the Institute of Pacific Relations, supported by wealthy patrons and dominated by communists, had great success in deceiving the U.S. government about the intentions of the Chicoms, cutting off aid to Chiang Kai-Shek after World War II, and setting the stage for the communist attack against South Korea in 1950. Bouscaren names many American persons and organizations which have aided the communists in the fields of education, labor, government, and the United Nations. He feels that loyal officials should learn how to identify such people and fronts and should take a firmer stand against their treasonable activities. He concludes that, if five percent of the American citizens joined to fight communism, the battle at home would be all but won. His concluding chapters tell how this can be done.

No less valuable than the book itself is the appendix. This contains a checklist for individual anti-communist action, a list of 55 American anti-communist organizations, and an extensive bibliography of books, government publications, and periodicals which deserve attention and support.

Those, who have sworn to bear true faith and allegiance to the United States of America and to serve them honestly and faithfully against all their enemies whomsoever, will find that *A Guide to Anti-Communist Action* is an indispensable handbook.

MODERN FOUNDRY PRACTICE

Third Edition
By E. D. Howard, Technical Editor
Published in 1959 by
Philosophical Library, Inc.
15 East 40th Street, New York 16, N.Y.
$15.00
460 pages of text and illustrations
Reviewed by
Frank H. Martin
Metallurgist with the Department of the Navy,
Bureau of Ships

This book is published as a guide to British foundry work, and is an ideal handbook for apprentices, students, craftsmen and executives in the industry, and for others wishing to obtain an understanding of foundry techniques.

The book covers all phases of foundry operation. The descriptions are based on British practice but the principles apply equally to that of other countries. The book contains thirteen chapters, prepared by specialists within their respective fields of foundry operation. The book is well illustrated and comprehensive, and is an excellent supplement to American publications on the subject.

Chapter headings;

1. Ferrous Metals
2. Heat Treatment in The Foundry
3. Non-Ferrous Alloys
4. Aluminum Alloys
5. Melting Furnaces
6. Pattern Equipment
7. Moulding Sands and Dressings
8. Moulds and Mould Making
9. Cores and Core Making
10. Moulding Machines
11. Modern Moulding Developments
12. Fettling Practice
13. Foundry Mechanization

The first four chapters deal with foundry metallurgy relative to various elements found in material compositions, heat treatments for mechanical properties, and methods of melting and refining compositions of ferrous and non-ferrous materials. Within these chapters the authors touch upon the causes of unsoundness frequently experienced in the casting of the materials under discussion.

In chapters Five through Ten, the authors have presented a comprehensive explanation of the manual and mechanical methods used in the foundry to produce castings. They cover the use of the main types of furnaces for melting ferrous and non-ferrous metals, and describe the wide range of pattern equipment and materials used in producing light, medium and heavy castings. The chapter on Sands and Dressings is a detailed account of the study of foundry sands, which constitutes one of the main avenues of foundry technology, sand testing being an essential part of foundry control. Chapter Nine describes the art of manual and mechanical core making. The author covers the field of resin bonded core sand, and describes the method of producing shell cores for shell moulding and cores made by the carbon dioxide (CO_2) process.

Chapter Eleven is devoted to two of the modern developments in the foundry field. The first is the Shell Moulding Process, often referred to as the "C" process, after Herr Croning, its German inventor. The shell moulding process originated in Germany during World War II and was introduced in the United States in 1948-1949. This chapter also describes the Carbon Dioxide Process, often refered to as the "CO_2" process. Both of these processes deal with the methods used in mould and core making.

Chapter Twelve is devoted to the cleaning and trimming of castings. The text deals with sand blasting, tumbling and various other requirements for preparing the castings for the finishing department, such as grinding, sawing, flame cutting and pickling.

Chapter Thirteen, the concluding chapter, is devoted to mechanization in the foundry. It deals with the various types of handling equipment available to the foundry; a short description of each type is given including conveyors, hoists, sand handling equipment, metal handling and pouring equipment.

THE ATOM AND THE ENERGY REVOLUTION

By Norman Lansdale
Published by Philosophical Library, Inc.
15 East 40th St., New York 16, N.Y.
$6.00 200 pages

Reviewed by
Maurice R. Hauschildt
Design Division, Bureau of Ships, Washington, D.C.

This book is for the businessman or technical man in fields related to atomic energy, who is interested in an overall non-technical view of atomic energy and its effect on the world. The author has revised a study previously made by Harold Whitehead and Partners Ltd., management consultants, on the implications of atomic energy in the world today and in the future. It is well illustrated with pictures of various uses of atomic energy. The political and economic effects of atomic energy are of chief interest to the author.

An assessment of current energy resources of all types is made and compared with present and future requirements for various countries and areas of the world. Various sources of power are discussed and evaluated including wind, water, vegetable, solar, electro-chemical electricity generation, and direct conversion of radiation to electricity.

The history and fundamentals of fission and fusion processes are presented and various methods of releasing atomic energy are described. A brief survey of location and availability of fissionable and related materials for nuclear power plants is given.

Various applications of atomic energy are discussed generally, including marine applications. He concludes that there is not much chance of nuclear powered merchant ships under 10,000 tons displacement. The radioactive and insurance risks in case of accident are emphasized. The observation that reactors are not well suited to continual load changes is open to question. Naval ships are evaluated as good but expensive applications particularly for submarine and high powered designs.

The development, history and potentials for atomic power are examined for many of the more important countries and areas of the world. The various political and commercial organizations, both national and international are discussed. The problems of radiation, insurance, finance and international control are well covered. The author concludes that the atomic energy implications of today demand the creation of a world political authority, which few national governments, if any at all, are at present willing even to contemplate. He believes that the exhaustion of conventional fuels in the near future, and the inability of atomic energy alone to replace them, may well make it a necessity to establish a world technical and political authority for exploitation of all forms of energy.

This book is of interest to the businessman, economist and citizen interested in world affairs, but it is likely to be disappointing to the technical man in fields related to atomic energy.

A HANDBOOK ON TORSIONAL VIBRATION

British Internal Combustion Engine Research Association
Edited by
E. J. Nestorides
(Cambridge University Press 1958, xxii + 664 pages, $19.50)

Reviewed by
Russell J. Bartell
Design Division, Bureau of Ships, Washington, D.C.

The increasing demand for high performance machinery magnifies the design problems associated with torsional vibration. The British Internal Combustion Engine Research Association (BICERA) is obviously well aware of this, and has made a timely contribution with the publication of this handbook.

The term "handbook" is somewhat of a misnomer if taken in the usual engineering sense. If "handbook" could be redefined as "a practical textbook," then this book fulfills the definition. The complex diversification of engineering problems requires more than a tabulation of "handy-dandy" relations.

The book is logically divided into four parts. The first part, entitled "Preliminary Calculations and

Measurements," deals with the determination of inertias, stiffnesses, and natural frequency calculations.

The usually annoying subject of crankshaft inertia and stiffness is well treated. The BICERA method of determining equivalent length of crankthrow is presented in detail. This is significant, since, according to BICERA, thirty-two engines of all sizes were torsiographed and crankthrow stiffnesses calculated using available formulae. The BICERA method of calculation was then applied and the results demonstrated that this method yielded a smaller range of error as well as a smaller average error. Twenty other methods of calculation of equivalent length of crankthrow are presented.

The section dealing with the calculation of natural frequencies is quite complete as it stands. However, it does not adequately cover the subject of electromechanical analogies. Most of the work performed today in the field of torsional vibration is accomplished by groups who can justify the use of computers. Obviously, a vibration man who is also well versed in electrical analogies can greatly expedite the solution of a vibration problem.

Part 2 brings to reality the results obtained in using Part 1. Part 2, "Evaluation and Prediction of Torsional Vibration Stresses," covers the establishment of dynamic forces, determination of stresses from vibration measurements, prediction of amplitudes and stresses, some torsional vibration codes, and cyclic speed variation.

Part 3, "Design and Operation of Various Devices for Limiting Vibration," in a sense describes the corrective action necessary to overcome the design difficulties that may be encountered in Parts 1 and 2. Damped and undamped tuning disks, various dampers and pendulum detuners are discussed in this part. The sections on "optimum" damping and untuned viscous-shear dampers are particularly well handled. The results of some fairly extensive investigations conducted by BICERA on the practical aspects of viscous-shear dampers are included. This work covers the characteristics of seismic masses and the design and construction of damper casings.

The fourth and final part of the book might be summed up with the old adage, "The proof of the pudding lies in the testing." Part 4, "Instrumentation," is not very complete, but it may be argued that a detailed description of instrumentation is beyond the scope of this book. It is felt, however, that a more complete bibliography could have been presented, particularly with regard to instrumentation work sponsored by the U.S. Navy.

The format of the book is pleasing, the units and nomenclature employed avoid the usual confusion, and the book is remarkably free of typographical errors.

THEORY OF BEAMS

The Application of the Laplace Transformation Method to Engineering Problems

by

T. Iwinski

Translated from the Polish by

E. P. Bernat

Published in 1958 by E. P. Bernat

Pergamon Press, Inc.

122 East 55th Street, New York 22, N.Y.

85 pages, $3.50

Reviewed by

R. Bart

David Taylor Model Basin

This monograph is the first part of a major project on the application of the Laplace Transformation Method to static problems in the theory of structures.

Some preliminary remarks on step functions and the properties of Laplace transforms are given, but without proof. The presentation is straightforward and neither presumes nor requires knowledge of mathematics beyond elementary calculus.

The basic differential equation of the elastic curve of a loaded beam is derived for general loading and then treated for a wide variety of end conditions but with constant rigidity modulus. The manipulation operations used to obtain solutions are essentially similar to the devices used to obtain solutions by the integration process normally used. The list of problems treated is exhaustive and extends even to obtaining the well-known Theorem of Three Moments.

The text concludes with a procedure for treating beams of varying rigidity moduli. Closed solutions are not obtained because such a problem is too general and depends on the particular variation.

The treatment is neither sufficiently rigorous for the mathematician nor sufficiently different in work required for the engineer, but is lucidly presented. The application of the Laplace Transformation Method to beams is interesting academically, but is

not sufficiently enticing when compared to the better known methods of structural analysis to attract a following. The adoption of the method must await the later portions of the project which presumably will deal with structural dynamics.

FLUID-DYNAMIC DRAG

by Sighard F. Hoerner Dr.-Ing.
Published in 1958 by the Author
148 Busteed Drive
Midland Park, New Jersey
400 pages of text and illustrations Price $15.00
Reviewed by
Capt. J. J. Stilwell, USN
Bureau of Ships, Navy Department

Chapter Headings:
Chapter 1—General Information
Chapter 2—Skin-Friction Drag
Chapter 3—Pressure Drag
Chapter 4—Wind Force on Structures
Chapter 5—Surface Irregularities
Chapter 6—Drag of Streamlined Shapes
Chapter 7—Drag due to Lift
Chapter 8—Interference Drag
Chapter 9—Internal-Flow Systems
Chapter 10—Hydrodynamic Drag
Chapter 11—Water-borne Craft
Chapter 12—Land-Borne Vehicles
Chapter 13—Aircraft Components
Chapter 14—Complete Aircraft
Chapter 15—Influence of Compressibility
Chapter 16—At Transonic Speeds
Chapter 17—At Supersonic Speeds
Chapter 18—Drag at High Speeds Affected by Viscosity and Rarefaction

"Fluid-Dynamic Drag" is without doubt the most detailed and complete compendium of theory, experimental facts, and practical engineering data to be found in aerodynamic or hydrodynamic literature. It is moreover unique in that the author has shared with his reader a vast amount of allied information and experience which has produced a very readable and at times fascinating technical "Cook's tour" through the practical applications of fluid mechanics.

Should you be interested in the wind forces on the Statue of Liberty, how to design a parachute, the best form for an automobile body, the resistance of a racing yacht, or the optimum shape for a space ship you will find it here, along with a host of other interesting and little known facts.

The work cannot be classed as a "handbook" as it demands of the reader sufficient engineering judgment and common sense to insure that he is applying the appropriate analysis to his problem and is not blindly picking up magic formulae or data that may appear similar. A better indexing and paragraph labeling system would be an improvement of the book as a reference text.

The book should be of particular interest to Naval Architects and Marine Engineers as it presents in useable form aerodynamic data and experience which can be applied to ships and appendages directly, as well as some refreshing new approaches to old resistance problems.

The six-hundred odd illustrations and charts and the liberal page references to source documents are most helpful to the investigator who may desire to dig deeply into any specific point raised in the text.

The international flavor of the book adds much to its interest. The author himself was educated in Germany, served as head of aerodynamic design in the Junkers A.G., later as research aerodynamicist at Messerschmitt A.G., and most recently is in the United States as a consulting specialist for hydrodynamics in a Naval Architectural firm. He has drawn data for his book from all over the world.

In short, this volume should be considered a "must" for anyone charged with estimating or improving the resistance (Drag) of anything subjected to motion through a fluid be it air or water.

DICTIONARY OF ASTRONOMY AND ASTRONAUTICS

By A. Spitz and Frank Gaynor
Published in 1959 by
Philosophical Library, Inc.
15 East 40th Street, New York, New York
$6.00 439 pages

Reviewed by
Leslie E. Alsager
Bureau of Ships, Design Division
Washington, D.C.

This book represents a much needed step in the right direction. The field of Astronautics as evidenced by the world's accent on space flight has been developing so fast in recent years that those who are new or very active in the field have had difficulty in keeping abreast with the many scientific terms which have come into usage. A dictionary or glossary of terms is, therefore, necessary.

This book does not follow the usual format commonly associated with a "dictionary," but rather follows the format of a "glossary." The authors have attempted through definition to set forth the new terms associated with Astronautics as well as those terms which are now in use in the specific field of Astronomy. Over 2200 terms are defined. While conciseness of definition is attempted, the authors, apparently in an effort to reduce the volume to a usable single reference book, have had to minimize the details and give general definitions in many cases. Throughout the work definitions are given in complete sentence form.

The authors do not claim to have produced a volume which will supplant any text book. Rather, they believe that it may provide answers in a ready form for many who do not have a complete technical library at their finger-tips.

The result of all of this work is to provide a handy tool for scientists and engineers entering the fields and for those others interested in the fields but not actively engaged in it. Scientists having an intimate knowledge of these may use this compilation as an easy reference for all sorts of technical terms which may not be found in the usual scientific dictionary. However, scientists having this intimate knowledge of the field would probably prefer to refer on specific subjects or terms to a scientific reference text or paper which gives more detail.

BOOKS RECEIVED

"Radioactivity Measuring Instruments," by M. C. Nokes, of the Atomic Energy Research Establishment, Harwell; published in 1958 by Philosophical Library, Inc., 15 East 40th Street, New York 16, N.Y.; 75 pages, $4.75.

"A Guide to Nuclear Energy," by R. F. K. Belchem; published in 1958 by Philosophical Library, Inc., 15 East 40th Street, New York 16, N.Y.; 77 pages, $3.75.

"Electrical Safety," by H. W. Swann; published in 1959 by Philosophical Library, Inc., 15 East 40th Street, New York 16, N.Y.; 292 pages, $15.00.

"The Radio Amateur's Handbook" (Thirty-sixth Edition—1959), by the Headquarters staff of the American Radio Relay League; published by the American Radio Relay League, Inc., West Hartford 7, Conn.; over 1300 illustrations, including some 500 tube-base diagrams. Price $3.50 in the United States, $4.00 in U. S. Possessions and Canada, $4.50 elsewhere.

"Helicopters and Autogyros of the World," by Paul Lambermont with Anthony Pirie; published in 1959 by Philosophical Library, Inc., 15 East 40th Street, New York 16, N.Y.; 255 pages plus 64 pages of illustrations; $10.00.

CHANGES IN MEMBERSHIP

ADDITIONS TO MEMBERSHIP

The Society announces with pleasure that the following have joined its ranks since the publication of the February 1959 issue of the JOURNAL.

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2. Single copy only is required.

3. Mathematical formulae used must be set up, preferably by hand printing, so that there can be no question as to symbols, subscripts and exponents.

4. Any drawing submitted must be black on white. A cloth or paper tracing is acceptable but not required.

5. Photographs may be negatives, but glossy prints are acceptable.

6. *Include on a separate page a short biographical sketch(es) of the author(s).* 50 to 100 words for each author is desired.

Manuscripts should be addressed to:

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We have a new supply of lapel buttons, in the form of the Society's official seal. These may be purchased by any member for fifty cents ($0.50) each.

By decision of the Council the official coloring of the Society seal is as it appears on the cover of this Journal. This is a reversal of the colors on the old lapel buttons. The size, one-half inch in diameter, remains the same.

Any member who wishes to exchange an old button for a new one may do so *free of charge*. Just send in the old button with your name and address.

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ASNE Financial Statements 1958

STATEMENT OF INCOME AND EXPENDITURES

EXPENDITURES

Publication:		
Printing	$25,884.65	
Engraving	2,719.83	
Manuscripts	5,296.83	
Postage	814.61	
Commission & Discount	2,991.06	
Exchange	1.20	$37,708.18
Salaries		20,916.56
General Expense		4,411.13
Awards		300.00
Rent		3,100.08
Bad Debts & Miscellaneous		7,042.60
Social Security		356.90
		$73,835.45

INCOME

Publication:			
Advertisements	$13,183.90		
Regular Dues	34,000.75		
Junior Dues	137.32		
Sales	540.57		
Subscriptions	19,141.91	$67,004.45	
Interest on Investments		1,295.29	
Banquet 1958		1,558.67	
Balance of Brand Award		20.00	
H. E. Yarnell Donation		100.00	69,978.41
NET LOSS			3,857.04

AMENDED BY ORDER OF 1958 AUDIT COMMITTEE:

Insurance item listed on Page 1 of this report has always been carried as an Asset at full value on Page 3. Since this item represents the full premium on a three year policy which terminates 7/30/59 and paid during 1956, the 1958 Audit Committee would not approve this as an Asset for 1958. They instructed that the entire value should be deducted from the Net Assets as of 12/31/58 and increase the Net Loss for 1958 by this amount ($113.45). The 1958 books have been so amended.

	113.45
AMENDED NET LOSS	$ 3,970.49

ASNE Financial Statements 1958

BALANCE STATEMENT

Assets January 1, 1958			$51,048.37
Assets December 31, 1958			
Cash		$19,222.98	
Petty Cash		97.62	
Accounts Receivable:			
Dues	$ 75.00		
Sales	70.25		
Subscriptions	403.00	548.25	
Investments:			
Equitable Savings & Loan	10,000.00		
First Federal Savings & Loan	5,000.00		
Liberty Savings & Loan	10,000.00		
Perpetual Building & Loan	10,000.00		
Washington Gas Light Co. Bonds	997.50		
U.S. Defense Bonds, Series K	5,000.00	40,997.50	
Furniture		2,114.24	
Manuscripts Paid in Advance		2,547.50	
Die & Medal for Annual Award		190.50	
Banquet Paid in Advance for 1959		123.57	
		$65,842.16	
Liabilities (Accounting Only)			
Advertisements Paid in Advance	161.70		
Dues Paid in Advance	751.25		
Subscriptions Paid in Advance	17,587.30		
A.G.F. Special Retirement Acct.	264.03	—18,764.28	
Net Assets December 31, 1958			47,077.88
NET LOSS			3,970.49

Audited and Found Correct 26 February 1959
s/GILBERT S. FRANKEL
s/E. H. BATCHELLER
s/D. E. REDMON

Respectfully submitted
s/J. E. HAMILTON, *Secretary-Treasurer*

Securities in Society's safe deposit box at American Security and Trust Company, Washington, D.C., examined and found as reported herein.
s/GILBERT S. FRANKEL

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(See reverse side for required qualifications for various classes of membership)

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I, _____ hereby apply for membership in the American Society of Naval Engineers, Inc., and enclose $10.00 as my annual membership dues for the year _____ $8.00 of which is for a subscription to the JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS, INC., for one year. I submit the following information:

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1926 **Commander A. M. Charlton, U. S. Navy**
1927-28 **Commander H. B. Hird, U. S. Navy**
1928 **Captain O. L. Cox, U. S. Navy**
1929-30 *Commander H. T. Smith, U. S. Navy*
1931 **Captain O. L. Cox, U. S. Navy**
1932 **Commander H. F. D. Davis, U. S. Navy**
1933-34 **Commander H. B. Hird, U. S. Navy**
1935-36 **Commander C. S. Gillette, U. S. Navy**
1936-38 **Commander Roger W. Paine, U. S. Navy**
1938-40 **Lt. Comdr. Guy Chadwick, U. S. Navy**
1940-44 **Captain J. E. Hamilton, U. S. Navy**
1945 **Commander R. T. Sutherland, Jr., U. S. Navy**
1945-48 **Captain F. W. Walton, U. S. Navy**
1948-51 **Captain J. E. Hamilton, U. S. Navy**

Past Assistant Secretaries

1939-40 **Lieutenant Milo B. Williams (C.C.) U. S. Navy**
1941-44 **Lt. Comdr. Robert T. Sutherland, Jr., U.S. Navy**
1945 **Captain F. W. Walton, U.S. Navy**
1945-47 **Commander Floyd B. Schultz, U.S. Navy**
1947-49 **Commander C. H. Meigs, U. S. Navy**
1949-50 **Commander Frank C. Jones, U. S. Navy**

Clerk and Administrative Assistant

1908-58 **Mr. Arthur G. Fessenden**
1958- **Miss Ruth M. Leonard**

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*Names in italics deceased